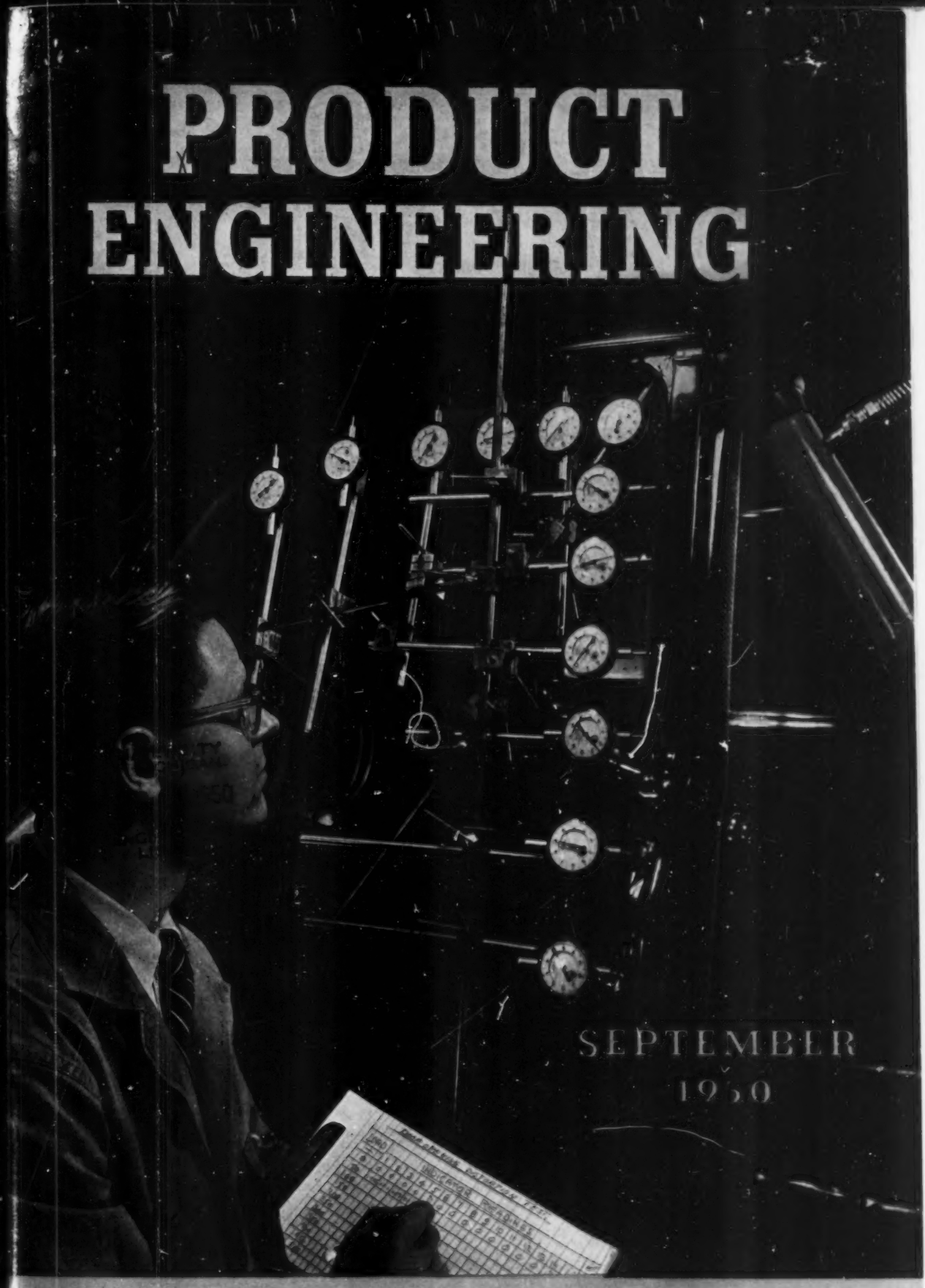
PRODUCT ENGINEERING



SEPTEMBER
1950

IN THIS ISSUE — Synthetic Hydraulic Fluids + Linear Motion Bearings + Planetary Gears + Electric Motor Duty Cycles + Fluid Magnetic Clutches + Screw Fasteners

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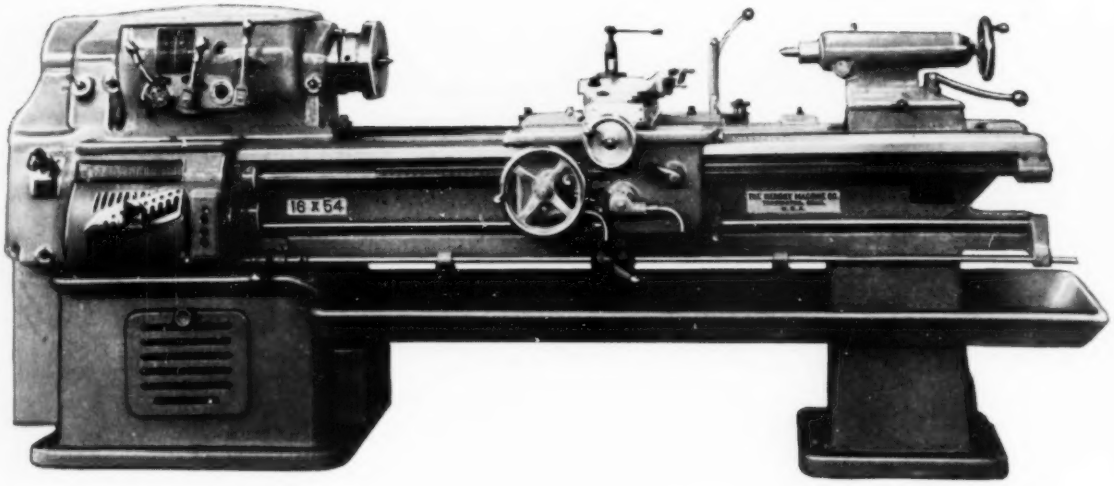

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VOLUME 21

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Are There Too Many Engineers?

TODAY THERE APPEARS TO BE an abundant supply of "engineers." Almost any 1950 engineering graduate will claim there is an oversupply.

According to figures furnished by the Office of Scientific Personnel of the National Research Council, the postwar spurt in college enrollments about balances the wartime losses. The same will soon be true of Ph.D. degrees. But numbers alone do not tell the whole story.

Less than 5 percent of those graduating with a Bachelor's degree have an I.Q. of 141 or better. This I.Q. figure is about the median for those holding a Doctor's degree. Less than 35 percent of the Bachelor degree graduates have an I.Q. of 125 or higher. Half of them have an I.Q. of less than 119 and one-fourth of them fall below 114. A high school graduate with an I.Q. of 125 is rated as good college material.

The foregoing explains why the lowest 25 percent of our engineering college graduates and many in the next 25 percent can be largely discounted in so far as responsible engineering design jobs are concerned. Because I.Q. ratings are not usually considered when hiring college graduates, many engineering design departments are cluttered with young college graduates who are hopeless to advance. Such men often stand in the way of a deserving, qualified and promising new engineering graduate getting a job.

But there is a place for the graduate "engineers" who do not have the necessary abilities to develop as designers. As pointed out by Mr. H. L. Ratchford, of Sylvania Electric Products, Inc., in a letter to the Editor presented on another page, there is "need for more engineers as foremen and supervisors . . . industry is requiring more fundamental know-how of engineering . . . more and better engineering control is paramount all along the line." To fill such positions does not require men who are exceptional in mathematics, physics, chemistry and related subjects.

Both colleges and business should distinguish between men who give promise of acquiring, or who have, merely an understanding of engineering and technical know-how and those men of high I.Q. and training who are, or will be, capable of handling and solving advanced fundamental scientific and engineering problems. In the former group belong the men who can qualify as sales engineers and production supervisors. In the latter group belong those to whom industry must look for the solutions of scientific and technical problems, design developments and the creation of new products.

Some colleges and universities have recognized that many of the young men seeking engineering degrees have qualifications for supervisory and management positions but are unable to master mathematics and physics sufficiently to qualify as design engineers. Hence, the courses called "Industrial Engineering", "Management Engineering", and by other names. A similar recognition that there is another large group of students that should not aspire to levels higher than supervisor or production manager would be a big step in raising the quality and decreasing the quantity of the mechanical and electrical engineering graduates.

With finer sifting at the college level, basic engineering courses could be broadened, requirements raised, and a better product delivered to industries.

The need today is not for more engineers, but for more better engineers. There are places for those who cannot hope to qualify in the upper category, but they should not be offered to industry as qualified mechanical or electrical engineers and thereby stand in the way of deserving men.

G. F. Rodenholt

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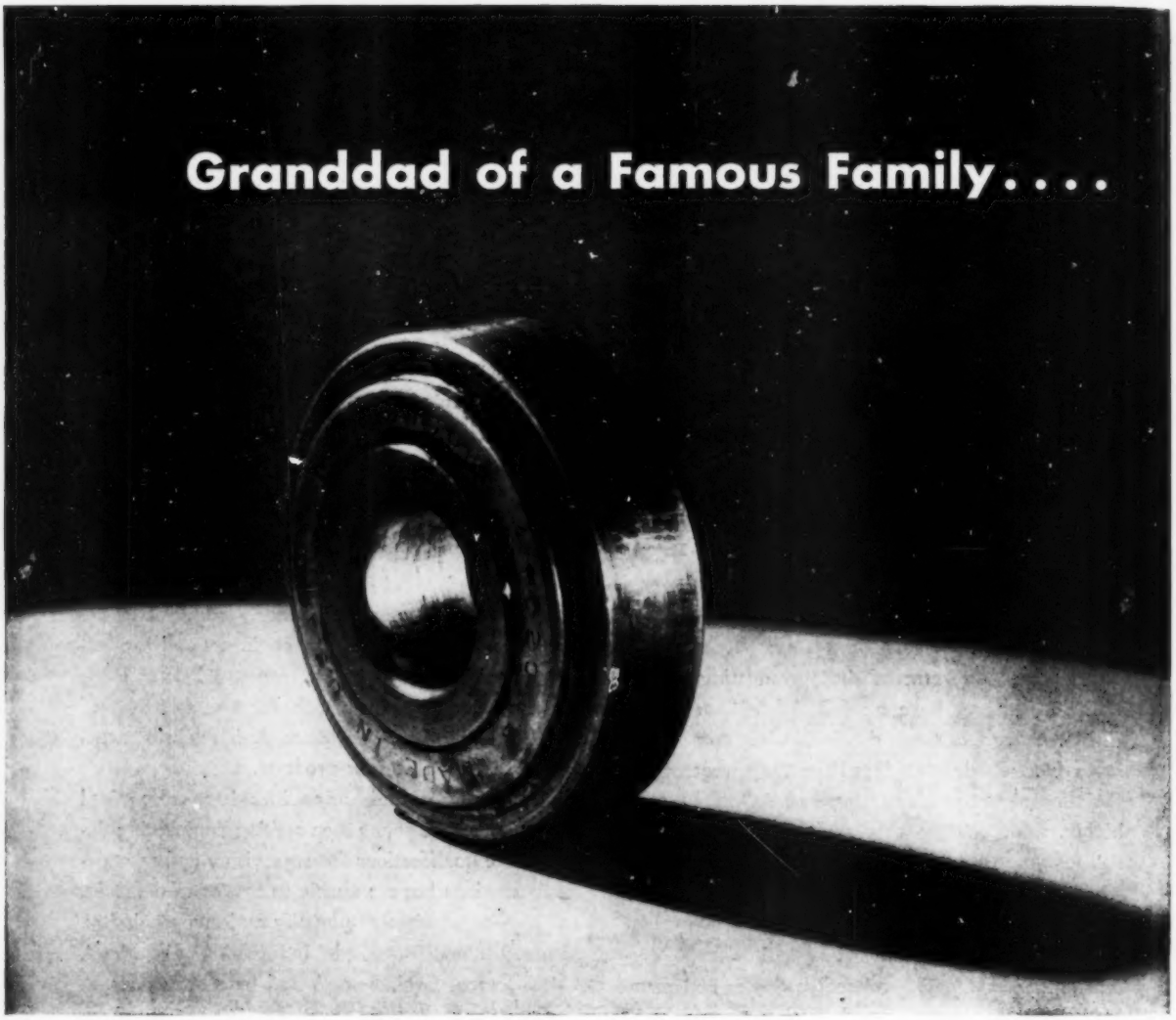


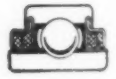
This self-sealed ball bearing, one of the earlier types originated by New Departure, was removed and examined in the twentieth year of its service in a grinder. No appreciable wear had occurred and the internal fit-up was still *well within* the tolerances of a new bearing.

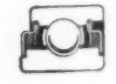
While this is eminently satisfactory, but not unusual service, it points up the significant fact that for assured performance, for the ability to eliminate periodic adjustments and to reduce lubricating or other maintenance to the simplest possible terms, the ball bearing is *outstanding*.

It was full knowledge of the distinctive advantages of the ball bearing that led New Departure to pioneer and develop the self-sealed bearing, including many lubricated-for-life types, some of which are illustrated at each side.

This hard worked "Granddaddy", progenitor of a famous family, thus started a trend that has required New Departure to produce more than 159 million self-sealed ball bearings to date.

Data covering ball bearing dimensions, application, lubrication and other subjects gladly sent upon request.





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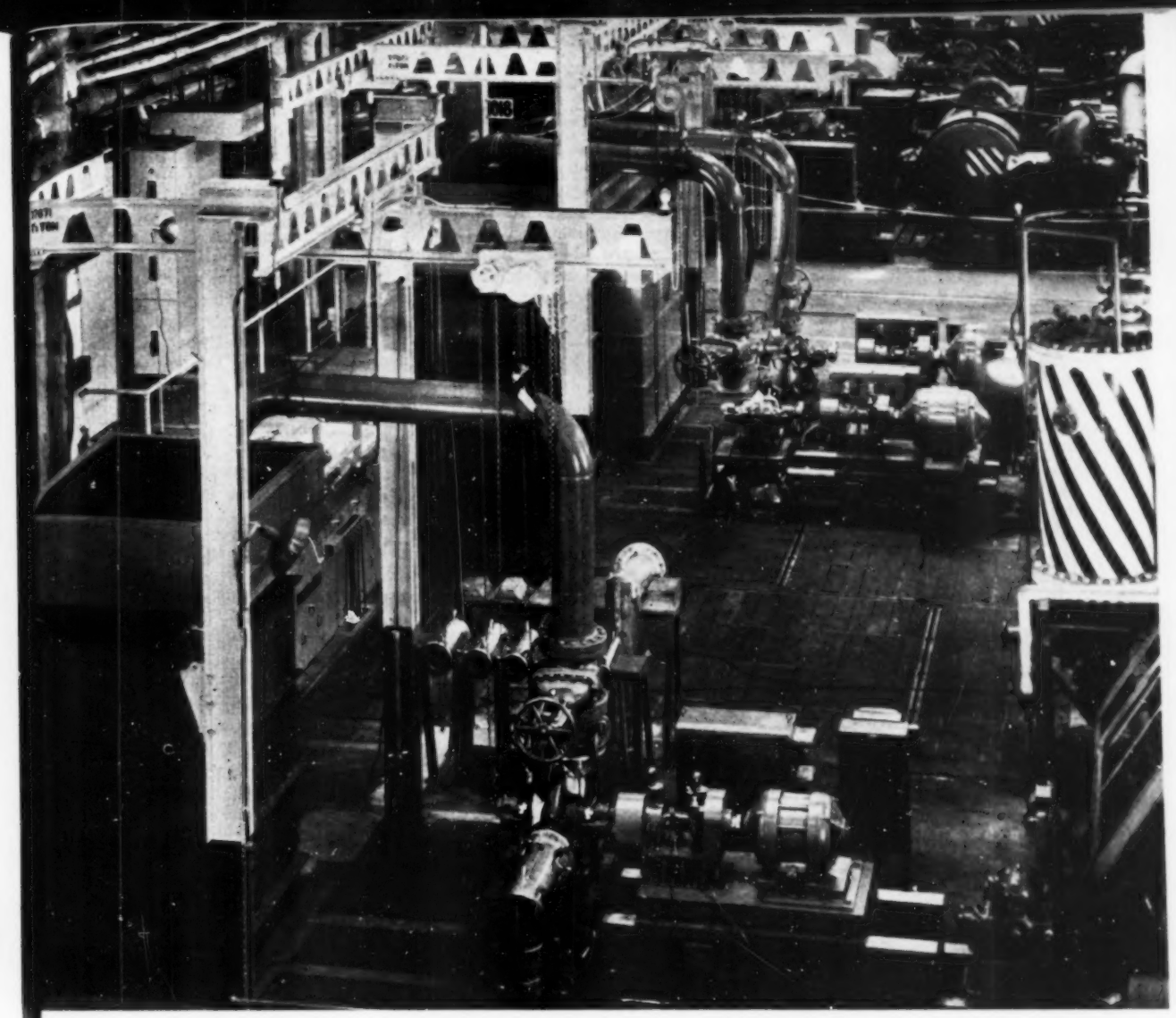

Fig. 1— Centrifugal pump test floor at Allis-Chalmers is designed to achieve low costs in testing; accurate test results; and maximum efficiency in use of personnel, equipment, and floor space.

Modernized Testing Facilities Reduce the Cost of Pumps

LLOYD M. DINGS, JR.
Special Engineer, Allis-Chalmers Manufacturing Company

TESTING THE PERFORMANCE of a product is often required to check or verify the fulfillment of engineering specifications. In addition, testing is desirable when test results yield data that can be used to develop and improve design. But testing costs must be included in the cost of the product. In any cost-cutting program, therefore, the cost of testing should receive as much consideration as the cost of materials, parts, and manufacturing.

Although frequently thought of as a shop operation, test procedure is closely associated with engineering objectives and the demands of customers. These objectives and demands determine in a large measure the capital investment in plant and equipment needed for testing and the cost of personnel for conducting the tests. The carrying charges of all these are a burden on the cost of the manufactured product.

A cost analysis, made by Allis-Chalmers of pumps of their manufacture, revealed that the charges for testing were a substantial part of the cost of producing pumps. On further study, it was concluded that lack of standard-

ization in test procedure, many little used items of test equipment, and inefficient use of floor space were responsible for high testing costs.

Following a decision to achieve lower costs, a completely new pump test floor was planned to obtain three objectives: (1) Reduced test costs, (2) improved accuracy in test results, and (3) maximum efficiency in use of personnel, equipment, and floor space.

The outcome was a new modernized testing laboratory, Fig. 1, which was occupied and put into use early in 1949. The results have fully justified the study and effort spent in planning. To date, the labor costs of testing pumps have been reduced by 50 percent, human error in obtaining data is no longer of consequence, and test facilities are installed safely in a limited floor area.

In this laboratory, pumps are tested to determine their operating characteristics and to obtain reference file data for further engineering development. Pumps are tested to determine capacity, suction lift, discharge head, efficiency, and horsepower required at rated speeds.

In planning the new test floor, a survey was conducted to learn of new methods that would yield the needed data in the shortest possible time.

A survey of industrial laboratories revealed that the recent developments in pump testing were usually variations of old methods. A few college laboratories were using some promising techniques, however, since these required considerable time to set up to obtain accurate data none was entirely suitable for industrial testing.

The second step in planning was to study the newest types of test equipment for their suitability in meeting the stated objectives. This survey indicated that by adopting or modifying some of the new types, and by designing some that were not available, the objectives could be met.

Capacity Measurement

Volume-rate of pump discharge formerly was measured at Allis-Chalmers with weirs, which because of their long approach flumes required considerable floor space.

Venturi meters were studied as a possible replacement for weirs. But further analysis indicated that Venturi meters were costly, difficult to read accurately, and required complicated pipe setups. In addition, a large amount of assorted sizes of pipe would be needed to make test connections. Time and labor costs of pipe fitting for test would also be high. Although initially a plan to use flow meters seemed promising, it was also abandoned. The difficulty in obtaining accurate readings because of "bouncing" of the instrument scale made it doubtful as to whether the laboratory should depend upon this type of instrument. The many sizes and types of pumps built, moreover, would add to the inventory problem.

Some laboratories and a few manufacturers had installations in which the weight of liquid is measured on scales. One type discharges water from the pump into a scale tank. The weight of water pumped in a measured time gives the pump capacity. But this system is complicated and awkward when used for large sizes and many varieties of pumps, principally because of the necessity to divert the stream of water from the tank after the elapsed time. However satisfactory such systems are for small installations, diverting a large stream of water requires too much complicated equipment to make it entirely satisfactory for large installations.

The final solution for measuring pump discharge was found in another field. In testing internal combustion engines, to measure fuel consumption the time required to pump a given amount of fuel from a scale tank is measured. In this setup a photo-electric cell and a light source are mounted on the face of a standard type dial scale. Two hands, or paddles, are fastened to the scale pointer shaft. The angular distance between the paddles is set to represent the weight of fuel to be consumed. As the weight of fuel in the scale tank decreases, the first paddle interrupts the light source and starts an electric timer. When the assigned arbitrary weight of fuel is consumed the second paddle turns into position, interrupts the light source again and causes the timer to stop. The method of measurement is extremely accurate, and an adaption of it was studied to solve the problem of selecting the best equipment to use in measuring accurately pump discharge.

For testing large capacity pumps, water would have to be pumped into the weigh tank, instead of out of it as in the fuel consumption tests. Because a pump must be kept in continuous operation during test, there are two alternatives as to the disposition of the water discharged by the pump: (1) The discharge could be directed into the weigh tank long enough to pump the test weight of water, and then diverted back into the suction pit; (2) or the water could flow through the weigh tank continuously and a means provided to close the tank long enough to weigh the required amount of water. Since diverting a large stream of water would require a costly mechanism, and would also create a serious splash problem, the latter alternative was adopted.

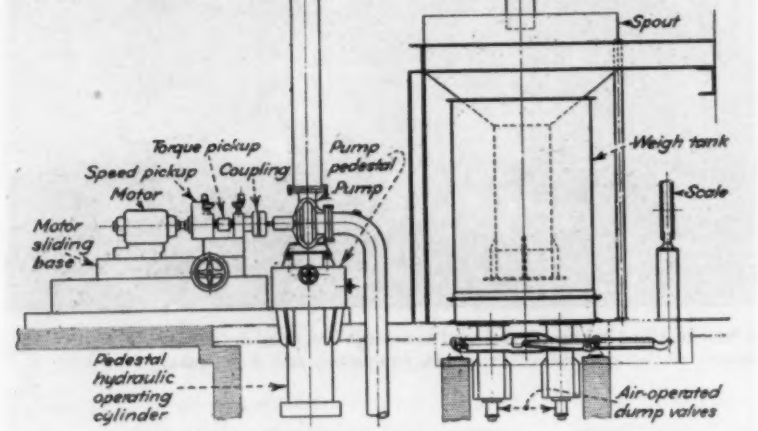

Fig. 2—New test for measuring pump discharge setup enables the operator to determine the time required to pump a known weight of water by merely pushing a button.

Designing The New Setup

From a survey of sizes and types of pumps built and sold, it was decided that three separate test stands were required; one for pumps up to 5,000 gpm and two for pumps up to

1,500 gpm. The weigh tanks were designed so that the length of time required to fill the tanks at the maximum design discharge capacity was 30 seconds. Thus the tank capacities were 2,500 and 750 gal respectively. The weigh tanks were fitted with dump valves to drain off the pump discharge into the pits when not taking a test point. The dump valves were of sufficient capacity to empty the weigh tanks at full pump discharge.

To prevent any whirling effects during drainage, three valves were placed in the bottom of each tank. Calculations indicated that three 12 in. valves on the large tank and three 10 in. valves on the small tanks would have sufficient capacity.

A new type of dump valve was designed to obtain simple and automatic remote control. The body of the new valve is a short length of pipe fastened to the bottom of the tank. The lower end of the pipe forms a seat against which a plate disk is pressed by an air cylinder to stop flow. The electronic scale circuit controls the valves.

The new test set up, Fig. 2, as modified from the system used to measure the fuel consumption of internal combustion engines operates as follows when testing pumps. Assume that water is constantly being discharged into the weigh tank with the dump valves open. The operator pushes a momentary contact button switch in the control room; thus energizing a holding relay and a pilot operated solenoid valve. The solenoid valve causes the air cylinders to close the dump valves. When the weight of water pumped into the tank reaches the tare weight, the first paddle on the scale dial interrupts the photo-electric cell light source to start an electric timer. When the weight of water pumped into the tank reaches the tare weight plus the test weight, the next paddle interrupts the light source to stop the timer and open the dump valves. Thus, merely by pushing a button the operator can determine the time required to pump a known weight of water.

Since the pumps tested cover a wide range of capacities, more than one test weight is required to obtain nearly uniform test times. The large scale is designed for two test weights and the small scales for three. Since a scale paddle is required for the tare weight and each test weight a counter is needed in the electronic circuit.

The electronic counter counts the impulses created by the interruption of the photo cell circuit. When it has counted the number of paddles for which it is set, the counter de-energizes the scale circuit, thus stopping the timer and opening the dump valves.

The downward thrust of the water being pumped into the weigh tank is received by a funnel suspended over the weigh tank from an independent structure. In the small end of the funnel is mounted a horizontal baffle. The falling water strikes the baffle and is deflected horizontally before falling into the weigh tank. Thus the downward thrust of the water is absorbed without affecting the scale reading This arrangement also affords some latitude in positioning the pump discharge pipe into large end of funnel.

Power Input Measurement

Before adopting the new procedure most of the pumps tested were driven by a calibrated electric motor.

A large inventory of calibrated motors was required to suit the horsepower and speed requirements of many different pumps. Before testing, each pump was aligned and coupled with a motor, on a set of parallels. The motor was then connected to a source of power by cables laid over the floor. Power input was measured with a wattmeter. Frequently it was necessary to use a potential or a current transformer in the circuit. Motor speed was measured with a tachometer. In addition to the costly inventory of motors, and a condition on the test floor that was somewhat dangerous, much time and labor was spent in setting up for the test.

The use of dynamometers for driving the pumps and measuring horsepower was also an expensive procedure; in addition, the apparatus occupied too much floor space.

In planning the new laboratory some study was given to the use of a Torquemeter for measuring power input. The use of a Torquemeter offered several advantages. It provides the basic measurement in foot-pounds; it is a small compact unit; it can be used over a wide range of speed; and its readings can be recorded easily.

To realize the inherent Torquemeter accuracy of 1/3 to 1 percent from 1/3 to full scale reading, the motor speed must be measured accurately, and a standard tachometer generator of such accuracy was not available. A tachometer unit, moreover, would make a gear shifting arrangement necessary because of the variable speed requirements. The entire speed record-

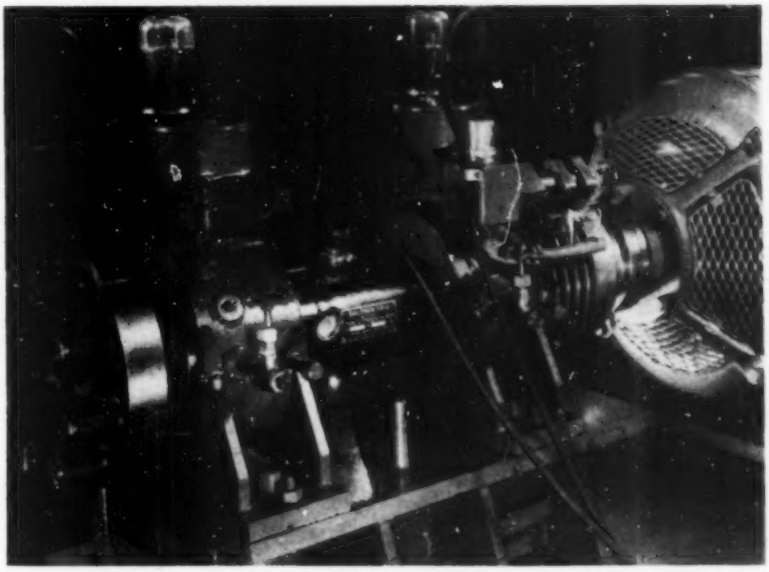

Fig. 3—Unit for measuring power input. Coupling on motor side incorporates frequency generator disks. The torque pickup is suspended between the bearing pedestals.

ing problem, therefore, was referred to the company electrical laboratory for solution.

After further study a unit, Fig. 3, was ultimately developed that is believed to be more accurate than any previous continuous speed recording unit. It consists of a tuning fork and a series of frequency generators, one for each required speed, mounted on a coupling at one end of the torque pickup shaft. The standard test speeds for the pumps are 600, 900, 1,200, 1,800 and 3,600 rpm. The number of poles on the generator disks are such that for each of the motor speeds a frequency of 180 cps is generated by one of the disks, when the motor is running at

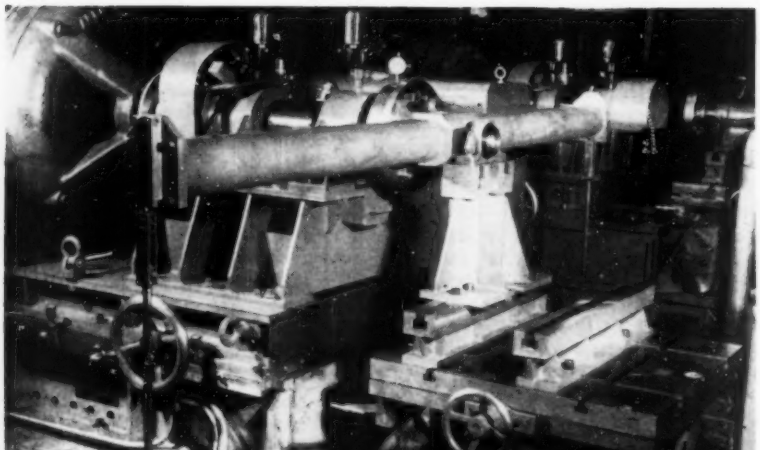

Fig. 4—Calibration beam and test weights arranged to calibrate torque pickups in place on the test stands in system for measuring power input to pump under test.

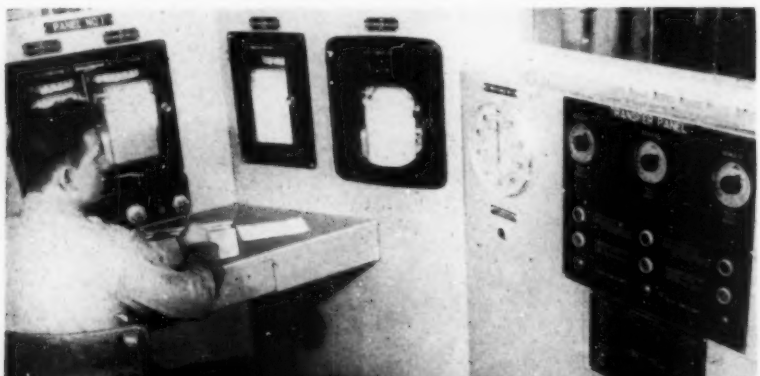

Fig. 5—All test readings are recorded and controlled in one central room. The transfer panel at the right contains the operating controls for the scales and the emergency stop buttons for the test motors. Data from any test stand can be read at either panel.

synchronous speed. The signal from an extremely accurate tuning fork having a frequency of 180 cps, is beat against the frequency from the generator on the torque pickup coupling. The motor, of course, drops below synchronous speed when it is loaded; therefore, the frequency generated is slightly less than 180 cps. The difference in frequency is read on a 20 to 30 cycle recording frequency meter and the converted to revolutions per minute. An accuracy of 1 in 3,600 rpm is obtained with the unit.

To achieve the strict alignment requirements imposed for the torque pickups (0.003 in. max runout), special torque pickup bearing pedestals were designed. The pickup is suspended between two bearing pedestals, each of which has a double high precision, labyrinth seal ball bearing. On each end of the torque pickup is a special pin and bushing coupling. An end clearance of approximately 0.010 in. is provided for the pickup to float within the couplings. One coupling incorporates the frequency generator disks. This construction reduces the amount of attention that must be given to pump and motor alignment, and also limits the force on the pickup to pure torsion. The friction losses encountered are small and easily cancelled out.

The torque pickups are calibrated in place on the test stands by means of a calibration beam and accurate test weights. After more than seven months of daily use it has not been necessary to readjust the torque pickup.

This system of power measurement has proven satisfactory, and makes it possible to record the test values on strip-type chart recorders on a central control panel.

Test Stands

Though the new methods adopted for obtaining pump capacity and power measurements did provide some time and labor savings, the greatest cost reduction possibilities were in setting up of a pump for test. Previously, standard pin and bushing couplings had to be keyed on the pump shaft. The pump and motor had to be blocked and shimmed on parallels to get correct coupling alignment. Motors had to be wired for each test. Pipe fittings had to be bolted. Piping often had to be made up especially for a test. Heavy piping had to be blocked to avoid overloading the pump flanges.

To make these operations unnecessary, hydraulically-operated permanent test fixtures were designed to couple the pumps to the test motors. Each fixture consists of a set of ways, upon which a movable table carries the test motor and the torque pickup bearing assembly. This unit can be moved horizontally by a screw turned by a hydraulic motor. The torque coupling can be inched ahead for final coupling, or it can be moved rapidly in traverse. The pump resting on parallels is bolted to a table, which is supported by an 18 in. dia hydraulic cylinder. With this arrangement, the pump can be raised, lowered and rotated to align the two coupling halves. Thus in a fraction of the time required by the old methods, one operator can now align a motor and pump by merely operating hydraulic controls.

Safety limit switches are so arranged that if the piston of the hydraulic cylinder should settle, or any of the fixture clamps release, the test motor is automatically stopped.

The use of tapered split bushings in the test couplings enables quick adaptation of one coupling to various shaft sizes. The couplings are fastened to the shafts with an Allen wrench. This coupling makes unnecessary the need for shimming couplings and for driving them off with a soft hammer.

On two of the test stands the motors and their wiring are permanent. On the third stand, one of two motors

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is needed to give the required driving speed. Even though exchanging the motors involves changing 21 leads, the changeover is made in 5 minutes or less by using quick clamping connectors.

To avoid the costly piping arrangement used in the past, permanent piping and adapters were designed. One 12 in. discharge pipe with gate valve attached is suspended, from a special cantilever bridge crane, over each test stand and weigh tank. Pumps are connected to the permanent discharge pipe by aluminum adapters with U-clamps. All suction pipe is fabricated of aluminum to reduce the weight on the pump flanges and to increase the ease of handling.

Water seal, gage, and priming lines are connected to the pump by self-sealing couplings. The prime advantages of these units are speed and ease of connection, also they can be closed off to prevent escape of water or air.

Control Room

The new pump test facility is rather unique in another respect. All test readings are recorded and controlled in one central control room, rather than being obtained from widely scattered test reading points.

The control unit, Fig. 5, comprises three sections, two identical recording control consoles and a transfer panel. Each recording panel has a 0 to 120, 0 to 250 and 0 to 500 ft pressure gage, a 0 to 30 ft vacuum gage, a 20-30 cycle frequency recorder for motor speed, and a torque recorder. The transfer panel enables the operator to transfer test readings from any one of the three test stands to either recording panel. The transfer panel also contains the operating controls for the scales, and the emergency stop buttons for test motors.

The only portion of the test not operated from the control office is the changing of the discharge valve opening. Since it is necessary to station an operator at the test stand for checking the mechanical operation of the pump, it was felt that he might as well operate the valve when necessary. This arrangement enabled great savings in cost by not using motorized valves.

Since all data are recorded, a permanent record is established from which test results can be studied at any time during design development projects.

New Alloys and Techniques Cut Cost of Powder Parts

RECENT REPORTS INDICATE that great strides are being made to put powder metallurgy on a commercial production basis. Significant progress is evident in the design of dies, presses and the production of special powders, all of which contribute to the reduction of production costs.

In some lines, such as oilless bearings and cemented carbide parts, metal powders have already proved commercially successful. Now the range of applications has been greatly broadened. New uses include such parts as bearings, bearing plates, odd shaped gears, clutches, ratchets and cams. Specialized powders combine copper with steel so that the copper melts when the steel is shaped to produce a self-brazing part; lead in copper to provide bearing properties; copper with carbon for brushes —the copper for conductivity and the carbon for an antifriction medium; and silver with tungsten in contact points—the silver for conductivity and the tungsten for heat and wear resistance. Various mixtures of copper, tin, iron, lead, graphite and silica are also being used to produce heavy-duty friction materials.

The Vanadium-Alloy Steel Company is concentrating on iron and steel powders that will sell competitively. To accomplish this, they are using a new process whereby powders are completely alloyed before delivery.

In essence, the method consists of making the desired alloy in a melt, then powdering a pencil-thin stream of the molten mass by blasting it with fine jets of water. The advantages claimed for such "prepared mixes" are: Little or no waste of material, minimum machining, ability to meet close tolerances at high production rates, control over porosity (useful in bearings and filters), and close control over strength through powder-mixing to a definite prescription.

Stainless-steel powder is already being tried out by one maker of fans as a sprayed-on covering for fan blades subject to corrosive vapors in the chemical industry. The same idea is being tested for filters in the chemical and food industries. Such applications would bring powdered metal into direct competition with clad steel and the all-stainless steel products. In another instance, an auto-engine builder is investigating the possibilities of using powdered alloy steels for making engine valves.

The American Electro Metal Corporation has two new types of stainless steel powders that can be sintered at low temperatures, have good moldability, and do not require high-purity atmospheres. One is a mix of 14 percent chromium, 86 percent iron; the other is iron plus 17 to 18 percent chromium and 8 to 9 percent nickel. Strength before sintering is said to be three to ten times higher than the strength of other commercial powders of similar composition.

Iron powders coated with soft metals have long been popular in the industry. Coatings such as copper, tin and nickel simplify molding procedures and add "green" strength. The Metals Refining Company has a method for obtaining such coatings, based on thermo-chemistry. Brittle iron powders coated this way can be pressed at normal pressures and then sintered at a temperature of about 2,000 Fahrenheit.

The S. K. Wellman Company is advocating clutches made of powder metals, claiming that such a clutch outperforms conventional types because of better heat absorption. Greater pressures and speed can thus be used in smaller more compact designs.

Metal powder is a natural for electrical and electronic equipment. Cores can be made by compacting in presses or by extrusion. The result, according to Standard Coil Products Corporation, is a significant reduction in high-frequency losses and the designer can specify a particular shape. Currently, hydrogen-produced iron powder is in favor for low-frequency cores, but magnetite powder, ferrites and flake powders are coming into more common use.

The most promising of these are the ferrites, which are oxides of metals instead of metal particles in an organic, insulating binder. The presence of the binder brings about a dilution of the permeability property of metal powders. In contrast, the oxides do not have to be powdered and diluted.

In addition to developing more and better metal powders, producers are also devoting much attention to cutting production costs. Typical of the courses of action being followed are: Improving the engineering of parts, redesigning dies and rearranging of production line components. A. S. Langhammer, who put the production of oilless bearings on a paying basis at the Amplex Manufacturing Company, has worked up detailed methods for efficient cost accounting of powder metal products.

Models Simplify Deflection Analysis of Curved Beams

A fast, accurate method for determining the deflection of curved beams with uniform or non-uniform cross sections and irregular arcs of curvature. Celluloid models, designed specifically to simulate the geometrical and loading conditions at the prototype beam, are used. Long, tedious calculations are unnecessary. Accuracy is within 1.5 percent in many cases. The authors tell you how it's done.

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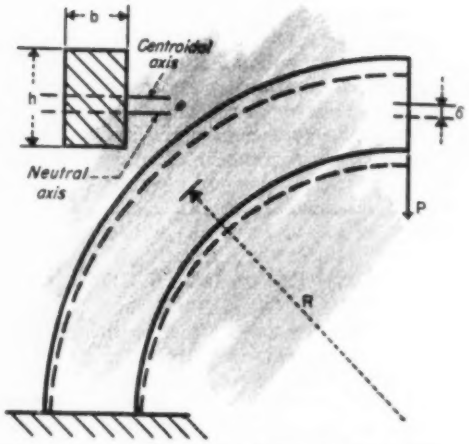

Fig. 1—Factors affecting the deflection of a curved beam are the loading, the radius of the centroidal axis, the area of the section, the moduli of elasticity and rigidity, and the distance between the neutral and the centroidal axis.

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OFTEN IT IS NECESSARY TO COMPUTE the deflections of curved beams that form part of the frame of presses, riveting machines, and many other types of industrial equipment. If such beams have a constant cross section and constant curvature, the calculation can easily be made. But if the cross-section is non-uniform or the arc of curvature changes at various intervals, the calculations become tedious and time-consuming. Celluloid models can be used to obviate this difficulty, provided that proper similitude is maintained between the model and the machine member.

Another advantage of model analysis is that deflections at various points along the neutral axis can be obtained with but little additional work, and an influence line for deflection at the loaded end can be drawn by using Maxwell's Reciprocal Theorem.[1]

The deflections of curved bars can be calculated by making use of Castigliano's Theorem, which states that "the derivative of the strain energy stored in a structure with respect to any load on the structure gives the deflection at the point of application of the load along its line of action." In mathematical terms,

$$\delta = \frac{dU}{dP}$$

where δ is the deflection, U the work of deformation, and P the force.

The total energy stored in a curved beam due to an external load applied in the plane of curvature may be expressed by:

$$U = \int_0^s \left[\frac{M^2}{2AEeR} + \frac{N^2}{2AE} - \frac{MN}{AER} + \frac{\alpha V^2}{2A_G} \right] ds \qquad (1)$$

in which R is the radius of the centroidal axis (radius of curvature); A is the area of the section in question; E and G are the modulus of elasticity and rigidity, respectively; α is a numerical constant determined by the shape of the section; M is the moment at the section; N is the normal force at the section; V is the shear force at the section; s is the length of the beam measured along the centroidal axis; and e is the distance between the neutral axis and the centroidal axis, Fig. 1, or

$$e = \frac{h^2}{12R}\left[1 + \frac{4}{15}\left(\frac{h}{2R}\right)^2\right] \qquad (2)$$

The first term in Eq (1) represents the energy due to bending. The second and third terms represent the energy due to the normal forces, whereas the last term represents the energy due to shear. In curved beams whose cross-sectional dimensions are small in comparison with the radius of

Table I—Design Dimensions for Prototype Curved Beams

Prototype Beam	b, in.	h, in.	I, in.^{4}	R, in.	R/h
A	1 1/2	12	216	36	3
B	1 1/2	12	216	24	2
C	1 1/2	12	216	12	1

curvature R, only the energy due to bending need be considered. On the other hand, if the cross-sectional dimensions are relatively large, the effect of shearing and normal force increases and therefore the energy due to these forces must be considered.

When Eq (1) is applied to the curved beam shown in Fig. 1, the deflection along the line of action of the load P, is

$$\delta = \frac{RP}{4AE}\left[\frac{R}{r} + \frac{\alpha E}{G} - 1\right] \qquad (3)$$

For a steel beam of uniform rectangular section of width b, depth h, and constant curvature R, $\alpha = 1.2$ and $E/G = 2.6$. Eq (3) then reduces to

$$\alpha = \frac{\pi PR}{4AE}\left[\frac{180R^4}{h^2(15R^2 + h^2)} + 2.12\right] \qquad (4)$$

The first term of Eq (4) gives the deflection caused by bending stresses, while the second term gives deflection caused by direct stress and shear. The deflection due to bending for various ratios of the curvature of a beam to its depth of section (R/h) can be computed and expressed as a percentage of the total deflection. As shown in Fig. 2, for values of R/h greater than 2, the deflection is not more than 3 or 4 percent in error if the effect of direct stress and shear is ignored.

Thus, as indicated by Eqs (1) and (3), a definite relationship can be established between a model and a full-sized curved beam if:

A—The beam and model are geometrically similar.

B—Their moments of inertia at corresponding sections are in constant ratio.

C—Their cross-sectional areas are in constant ratio.

D—Poisson's ratio is identical for the model and its prototype.

Frequently not all of these conditions can be satisfied. If the portion of the deflection due to direct stress and shear is negligible, only conditions A and B need be met. If an arbitrary geometric scale reduction is selected for conditions A and C, there are narrow limits within which condition B is also met, but if met, frequently the model is of poor proportions. It is essential that the geometric scale reduction apply to both R and h.

Theoretical vs Model Analysis

To actually determine the relative importance of the various factors governing the accuracy of model analysis, three curved beams A, B, and C, Table I, of constant rectangular section with a 90 deg central angle were represented by five curved model beams a, b, c, d and e, Table II. All model beams and the two straight beams f and g were 1.0 in. wide, $\frac{1}{8}$ in. thick, and had essentially the same elastic properties even though Young's Modulus is slightly dependent on sheet direction.

Each prototype was therefore represented by five different models, each to a different scale reduction for R and h. Only in beam A and model e were the R/h ratios identical, so that the scale reductions were incorrect for both R and h in all other models.

Table II—Design Dimensions for Model Beams

Model Beam	b, in.	h, in.	R, in.	R/h
a	1/8	1	18	18
b	1/8	1	15	15
c	1/8	1	12	12
d	1/8	1	6	6
e	1/8	1	3	3
f	1/8	1	.....	.....
g	1/8	1	.....	.....

Since celluloid creeps and Young's Modulus is a function of the duration of stress, a simple load deflection curve for each model beam cannot be obtained. One of two other methods must be followed:

In the first method, the deflection of the curved beam must be measured at definite time intervals for which corresponding values of Young's modulus for the plastic are known. In the second method, the deflection of a constant section straight cantilever beam and the curved beam are determined: (a) simultaneously under the same force, or (b) through the use of a creep-compensating dynanometer

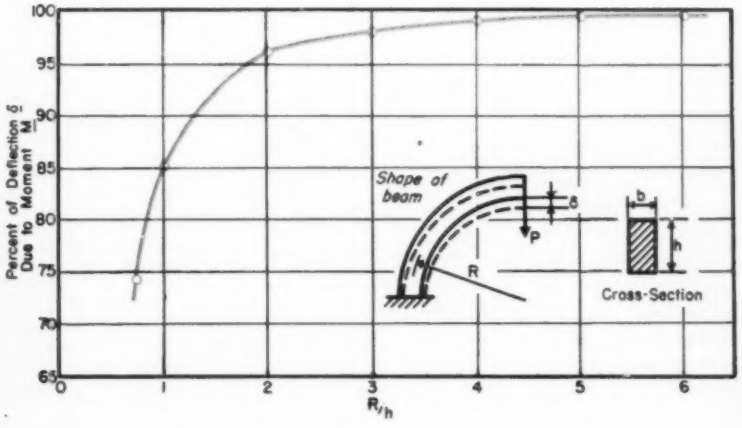


Fig. 2—How deflection of a curved beam varies with the ratio of the radius of curvature to the height of the section, R/h. For ratio values over 2, direct shear and stress has little effect on deflection, it not being over 3 or 4 percent in error if they are ignored.

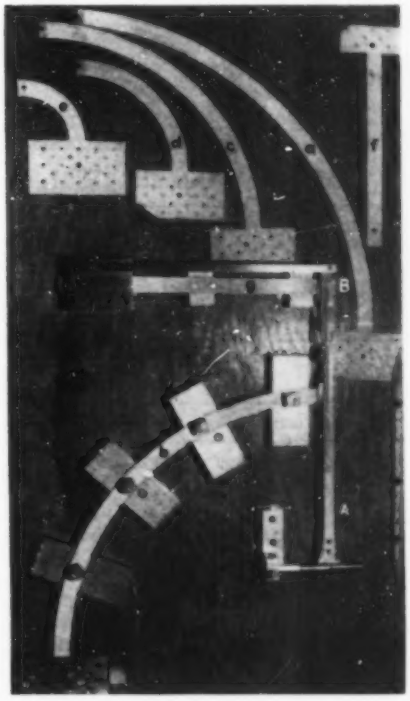


Fig. 3—Set-up for simultaneously determining relative deflections of curved and cantilever beams under the same loading. These ratios are used in model analysis. Other methods of compensating for creep are described in the text.

such as a celluloid loading spring.[3] Method 2 (a) was used in this case.

The set-up is illustrated in Fig. 3, which shows curved beam being deflected by a turnbuckle attached to the end of the cantilever loading beam. The curved beam is rigidly fixed at its lower left end so that as the turnbuckle is shortened, its right end moves up and to the left. A scale slides along a rod to follow this movement and record the vertical deflection. The cantilever loading beam is so clamped at its left end that translation without rotation to the left or right can take place. A special scale arm is attached to this base and fitted with a scale that records the true vertical deflections of the loading beam. By adjusting the turnbuckle, the curved beam can be made to deflect towards or away from the loading beam. Each adjustment of the turnbuckle requires positioning the cantilever beam so that vertical loads normal to the axis of the loading beam are simulated. Both scales are graduated in 1/100 in. and capable of being read to within ±0.002 in. with a watchmaker's loupe.

In testing each curved beam, the unstressed normal position of the curved beam was noted. The turnbuckle was then adjusted so as to produce equal deflections of the curved beam up and down from its normal position. Total deflections induced in the curved beams ranged from a maximum of 1.194 in. for beam *a*, Table II, to 0.200 in. for beam *e*. For each beam, the deflections of the curved and cantilever beams were proportional for the full range of deflection induced.

The relative deflection of the cantilever and curved beams as determined experimentally are given in Table III. These average ratios can be used to compare the calculated deflections of curved machine beams with those obtained experimentally with a model. The method that is followed makes use of the relationship that exists between the model cantilever and curved beam and an imaginary prototype cantilever and the prototype beam.[8] In this method the effect of shear on the cantilever beam is neglected. The theoretical deflections of the prototype beams *A*, *B* and *C*, as calculated by means of formula (4), are given in Table IV.

To illustrate the method, assume that a comparison is to be made between beam *A* and model *e*. Table III gives 2.000 as the experimental average ratio of cantilever deflection to curved beam deflection, for the above beams. This means that identical forces deflect the cantilever 2.000 in. and the curved beam 1.000 in. Consider now an imaginary prototype cantilever beam that simulates the model cantilever to the same linear scale reduction. The length of this cantilever is 5 x 12 in. = 60 in., since the linear scale reduction is 12 for both *R* and *h*; and 5 in. is the length of the loading beam. The force that will deflect this prototype cantilever 2.000 in. will deflect the prototype curved beam 1.000 in.

This imaginary prototype cantilever has a moment of inertia of 216 in.[4] corresponding to the moment of inertia of the curved prototype. (In both curved and straight beams, a width of one inch was chosen to represent an *I* of 216).

The force required to produce a 2.000 in deflection of the imaginary prototype cantilever, when the force acts at the free end and can be computed from the expression:

$$\Delta = PL^3/3EI \quad , \text{ and}$$

$$P = \frac{3\,\Delta\, EI}{L^3} =$$

$$\frac{3 \times 2.000 \times 30 \times 10^6 \times 216}{60^3} = 180 \text{ kips.}$$

The theoretical force *P* to deflect beam *A* by 1.000 in. is, according to Table VI, 175 kips, a value close to that found by model.

The lengths of all such prototype cantilevers used in calculating the loads corresponding to a deflection of 1.000 in. of the prototype beams are shown in Table V. These lengths are based on the linear scale reduction of *R* of each model and on the lengths of the cantilever loading beams used in the test. The latter are 11 in. for *a* to *d*, and 5 in. for *e*.

In the example there is complete similitude between the beam and the model, but the deflection can still be determined with a model if such is not the case. For beam *C* and model *a*, the geometric scale reduction for *R* is ⅔ and for *h* is 12. Therefore, the first rule of similitude is not met. If the same calculations as for beams *A* and *e* are made, the theoretical load to deflect *C* by 1.000 in. is found to

Table III—Relative Deflections of Curved and Cantilever Model Beams

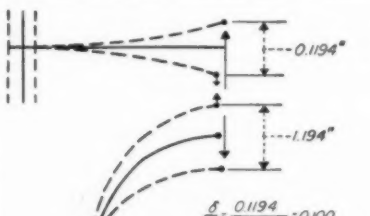
Model Beam	Ratio of Cantilever to Curved Beam Deflection	Example (Beam *a*)
a	0.100	
b	0.171	
c	0.340	
d	2.620	
e	2.000	

Table IV—Theoretical Deflections of Prototype Curved Beams

Prototype Beams	*R*	*R/h*	Due to *M*, *N* and *V*	Due to *M*
A	36	3	$5.72 \times 10^{-6} P$	$5.61 \times 10^{-6} P$
B	24	2	$1.72 \times 10^{-6} P$	$1.65 \times 10^{-6} P$
C	12	1	$0.233 \times 10^{-6} P$	$0.197 \times 10^{-6} P$

Table V—Prototype Cantilevers For Establishing Deflection Relationships

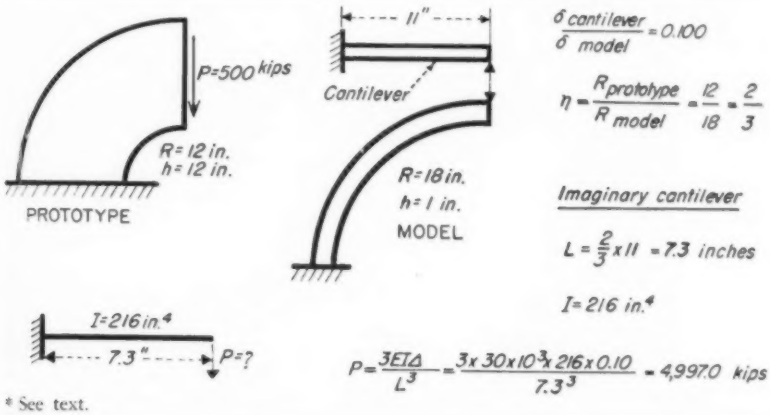
Lengths of Imaginary Cantilevers				Example:
Prototype Beam / Model Beam	*A*	*B*	*C*	Consider Prototype
a	22.0"	14.7"	7.3"	
b	26.4	17.6	8.8	*C* and Model *b*
c	33.0	22.0	11.0	$n = R_c/R_b = 12/15 = 0.8$
d	66.0	44.0	22.0	$L = n \times$ length of loading cantilever
e	60.0	40.0	20.0	$L = 0.8 \times 11.0 = 8.8$ See Tables I and II

Table VI—Comparison of Experimental and Calculated Deflections for Prototype Curved Beams

Model Beam			Beam A ($R/h = 3$)			Beam B ($R/h = 2$)			Beam C ($R/h = 1$)		
			Load for 1.0" Deflection		Percent Error	Load for 1.0" Deflection		Percent Error	Load for 1.0" Deflection		Percent Error
No.	R	R/h	Model	Theory		Model	Theory		Model	Theory	
a	18	18	182.6	178.0[b]	+2.6	612.0	606.0[b]	+1.0	4,997.0	5,076.1[b]	−1.6
b	15	15	180.7	178.0	+1.5	609.7	606.0	+0.6	4,877.8	5,076.1	−3.9
c	12	12	183.9	178.0	+3.3	620.7	606.0	+2.4	4,967.4	5,076.1	−2.1
d	6	6	178.1[a]	178.0	+0.1	600.5[a]	606.0	−1.0	4,807.0[a]	5,076.1	−5.3
e	3	3	180.0	175.0[c]	+2.9	621.2[d]	606.0	+2.5	4,969.0[d]	5,076.1	−2.1

[a] Total Force ÷ 0.995. 177.2 ÷ 0.995 = 178.1 (Factor 0.995 from Fig. 2).
[b] Load for 1.000 in. deflection due to moment only.
[c] Load for 1.000 in. deflection due to moment, shear and normal forces.
$\delta_{MNV} = 5.72 \times 10^{-3} P$; complete similarity between prototype A and Model e.
[d] Total Force ÷ 0.978: i. e. 607.5 ÷ 0.978 = 621.2 (Factor 0.978 from Fig. 2).

Table VII—Calculation of the Loading of a Machine Beam*



* See text.

be 4291.8 kips. Checking this load with model *a*, it is found that the load P is 4997.0 kips, which is 16.4 percent higher than the theoretical. Note that beam C has an R/h value of 1, and model *a* has an R/h value of 18. Furthermore, Fig. 2 shows that when R/h equals 18, the deflection due to direct stress and shear is negligible. When R/h is equal to 1, only 84.7 percent of the deflection is due to moment. Therefore, when comparisons are made between beam C and model *a*, the deflection in C caused by the moment only should be considered. Table IV gives $\delta = 0.197 \times 10^{-6} P$, when δ is due to moment only. Therefore, when $\delta = 1.000$ in., $P = 5076.1$ kips. The experimental load $P = 4997.0$ kips, is 1.6 percent smaller than the theoretical.

The experimental and theoretical forces in kips required to deflect beams A, B and C 1.000 in. are shown in Table VI. In preparing this table, due consideration was given to the R/h ratios of the models and prototypes. Thus, it can be seen that the errors are relatively small for a model analysis of this kind.

Examples of Model Analysis

The following are two examples illustrating the use of the model method.

EXAMPLE 1—To find the deflection of curved beam C with model *a* when C is acted on by a force P equal to 500 kips. This is the force required to deflect the prototype by 1.000 in. and is due solely to moment. Calculation of the machine beam in loading is given in Table VII. Since only 84.7 percent of the deflection in the beam is due to moment, the force required to deflect the machine beam by 1.000 in. is $4997.0 \times 0.847 = 4232.4$ kips, and

$$\delta = \frac{500}{4232.4} = 0.1181 \text{ in.}$$

The theoretical deflection is found to be:

$$\delta = 0.233 \times 10^{-6} \times 500 = 0.1165 \text{ in.}$$

EXAMPLE 2—With a beam similar to A, $P = 500$ kips, and model d, find the deflection due to the load.

$$L = \frac{36}{6} \times 11 = 66 \text{ in;}$$

$$\frac{\delta \text{ cantilever}}{\delta \text{ model beam}} = 2.62$$

$$\therefore P = \frac{3 \times 30 \times 10^6 \times 216 \times 2.62}{66^3} = 177.2 \text{ kips}$$

For Machine Beam: $R/h = 3$; deflection due to moment $= 0.978 \times \delta$.
For Model: $R/h = 6$; deflection due to moment $= 0.955 \times \delta$.

Therefore the load to produce 1.000 in. deflection in the full-sized beam is

$$P = 177.2 \times \frac{0.978}{0.995} = 174.2 \text{ kips}$$

The deflection due to 500 kips is:

$$\delta = \frac{500}{174.2} = 2.87 \text{ in.}$$

The theoretical deflection is:

$$\delta = \frac{500}{175} = 2.86 \text{ in.}$$

Thus it can be seen that the use of models gives sufficiently accurate results for most curved beam deflection analyses, and, as previously mentioned, is less tedious and time-consuming, especially if the beam is non-uniform in cross section or curvature.

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WHAT'S NEW IN METALS—Research on metals has been stepped up considerably the past two or three years. New light metal alloys, new uses for molybdenum and titanium, and recent developments in ferrous metals are of special significance to designers. These and other developments will be highlighted in the October issue of PRODUCT ENGINEERING, which will also contain a special full-color report on coatings for gray iron, a full compliment of design application articles on mechanical, electrical, and hydraulic equipment, and news of the metal show.

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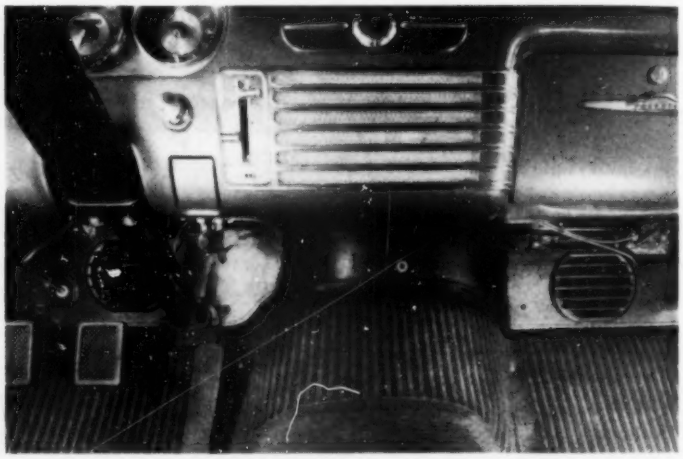
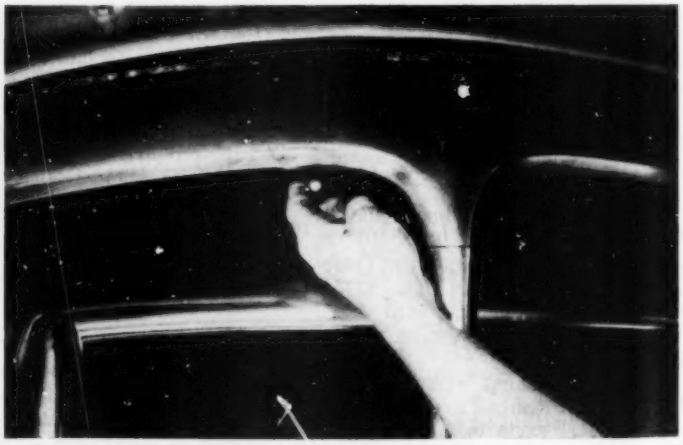
Optional Heads Simplify Engine Production

Body design of the Packard 1951 line of automobiles is completely new. Greatly increased glass area, lower hood and extended rear deck are the main styling changes. The cars are lower, longer and have trunk space almost double that of last year. Rear seats of some models are almost 12 in. wider. As in the past, Packard offers three series of models. Series "200" has a 122 in. wheelbase while the "300" and "400" series are on a 127 in. wheelbase. The adoption of high compression engine design has resulted in production economies, while increasing the number of power plants from which the customer can choose. Formerly, there were three Packard engines of 288, 327 and 356 cu. in. piston displacement. All engines had a 7 to 1 compression ratio. The largest of these has been discontinued, and the two remaining are available with either standard or high compression heads. Thus, there are four power plant combinations developing 135, 138, 150 and 155 hp at 3,600 rpm. The compression ratio of the 155 hp engine is 7.8 to 1, which is claimed to be the highest in the auto industry for full-size cars. Prominent interior styling detail is the use of biscuit-design, leather-like vinyl plastic in the back of the front seat, harmonizing with fabrics used on cushions and door trim. For maximum engine quietness, the Patrician "400" series cars are equipped with asbestos lined mufflers. This series also is equipped with a wrap-around rear window, while all series feature a one-piece curved windshield. Defroster outlets run the entire width of the windshield. The instrument panel has been simplified by the substitution of red warning lights for the ammeter and oil pressure indicator.

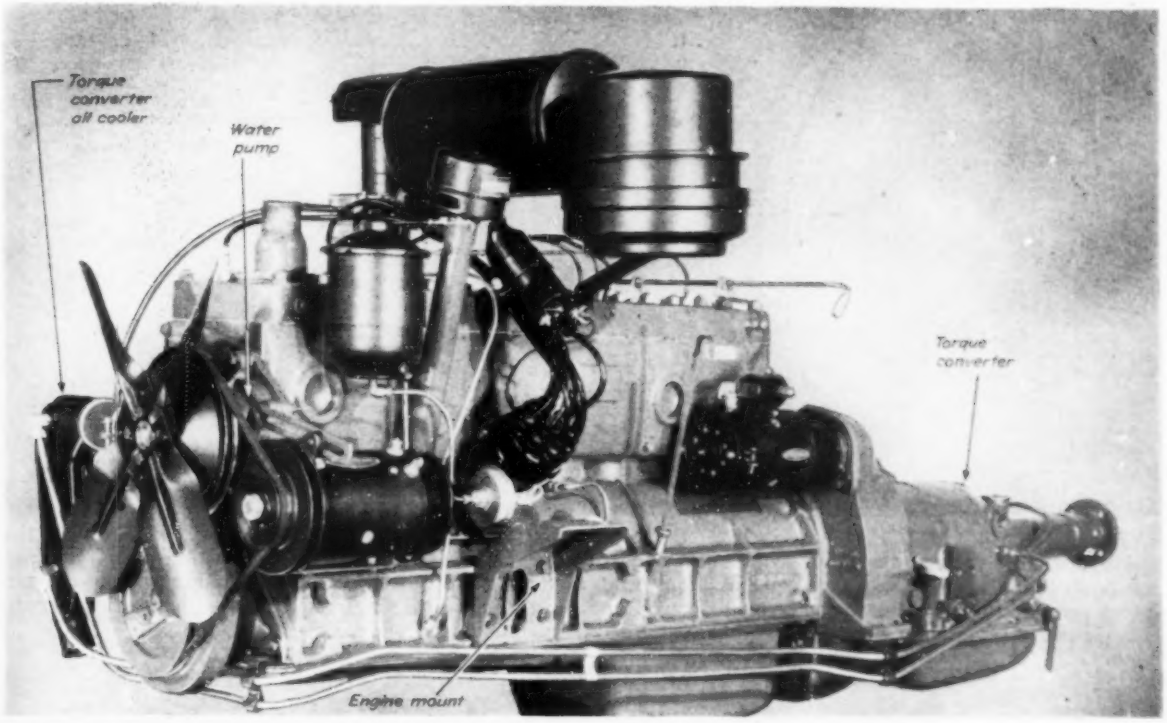
GLASS AREA is 26 percent greater than that of previous models. The Patrician "400" series, above, features wrap-around rear window.



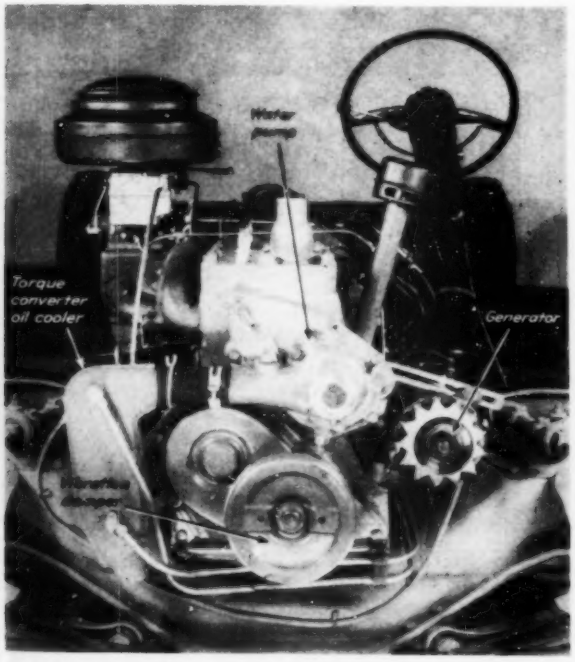
INTERIOR CHANGES include drawer-type glove compartment and red warning lights replacing battery charge and oil pressure meters.



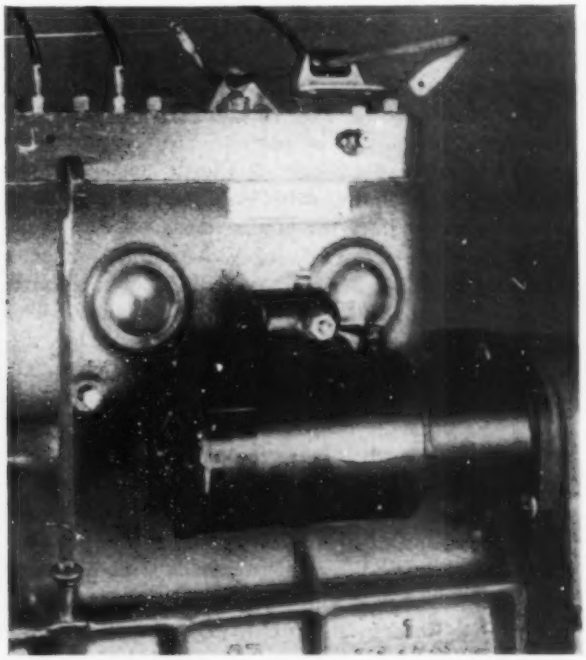
HOOD LOCK RELEASE is accessible through front grille. Hood hinges are at the back rather than at the sides as previously. Grille retains traditional Packard design theme.



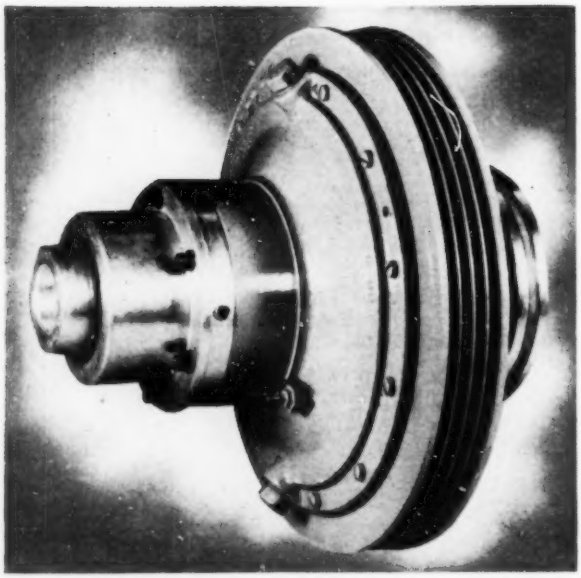
OPTIONAL HEAD raises compression ratio of the larger engine to 7.8 to 1 and boosts power to 155 hp. These figures compare with 7.0 to 1 compression ratio and 150 hp for standard engine. The torque converter oil cooler is now mounted directly on the engine to eliminate need for flexible tubing between engine and chassis.



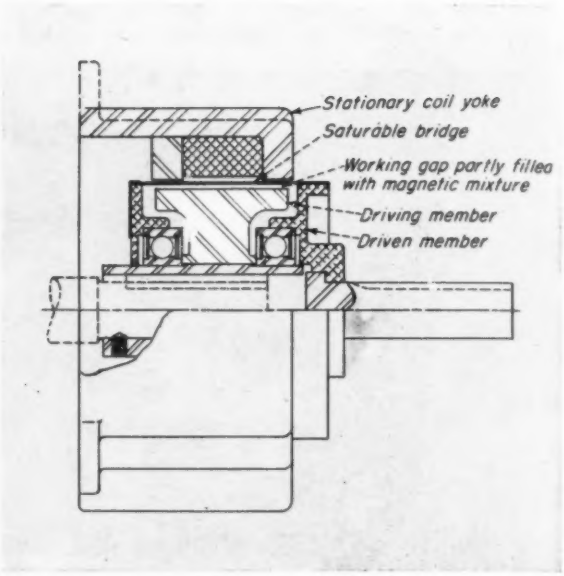
WATER PUMP has been moved down to allow lower hood and cowl. New location assure satisfactory operation of pump even when cooling system water level is low.



DIP-STICK PORT is now above engine ledge to lessen possibility of entry of dirt when oil level is checked. Starter is solenoid actuated and is mounted high for easy servicing.



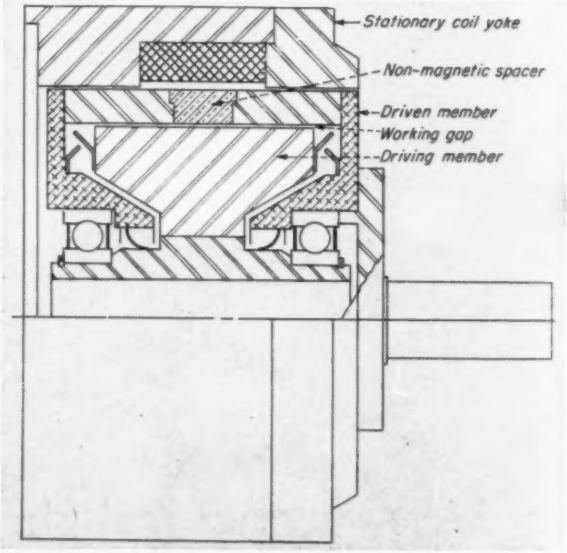
TYPICAL MAGNECLUTCH is for use between the motor and generator of a 5 kw power supply set. The clutch operates under continuous slip, with the degree of slip depending on generator load and the frequency of the power supply. Output is kept within one-half of one percent.



CROSS SECTION DRAWING, above, shows construction of typical Magneclutch illustrated at left. The cylindrical working gap is not quite full of the magnetic powder mixture. Centrifugal flingers help to keep the magnetic powder mixture away from the central bearing seals.

Quantity Produced Magnetic Clutches

Since the first magnetic-fluid clutch was announced by the Bureau of Standards in 1948 several small servo-type clutches have been marketed. Now, after an extensive development program, Vickers Electric Div., Vickers, Inc., is producing magnetic clutches capable of controlling large amounts of power. Vickers prefers the generic term "Magnetic-particle Clutch" instead of the usual "Magnetic fluid Clutch", since their product uses a dry magnetic mixture for power transmission. Tradenamed the "Magneclutch", the Vickers unit contains a mixture of iron powder and very fine flake graphite instead of the usual oil and iron powder mixture. This dry mixture makes the sealing problem much less difficult. Stationary joints are sealed with rubber "O" rings and a system of centrifugal flingers keep the mixture away from the central shaft seals. The simplest type of Magneclutch is made for direct on-and-off service. In this type of duty it has important advantages of remote control, smooth starting, control of acceleration, cycling duty, and small control signal. It is particularly useful for starting heavy inertia loads with minimum surge from the power supply. Since the maximum torque is established by the current coil, the clutch automatically operates as an overload safety device. It is especially good as a slipping clutch, and may also be used as a magnetic brake or dynamometer.



CROSS-SECTION SHOWS a differently designed Magneclutch. This smaller model is for fractional hp applications. There are a great many possible variations in possible clutch design. Double clutches or double exitation members can be used to give reversing or "Servo" duty. Also, since the coil power is very small, the reversing clutch makes an effective power amplifier. Size does not seem to be limited as long as the clutch has sufficient heat dissipating area.

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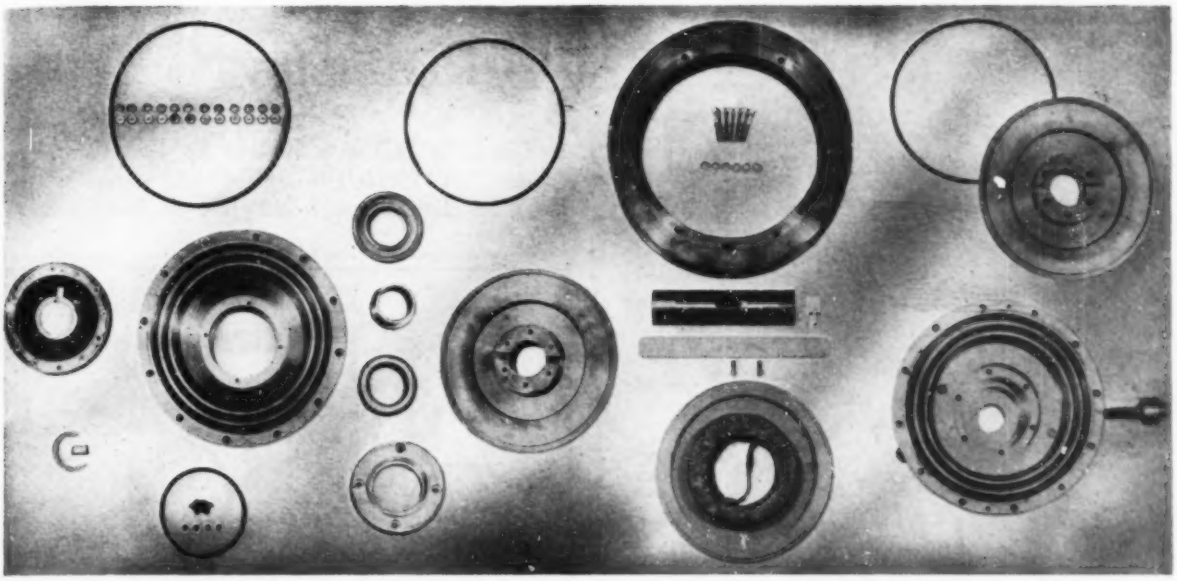
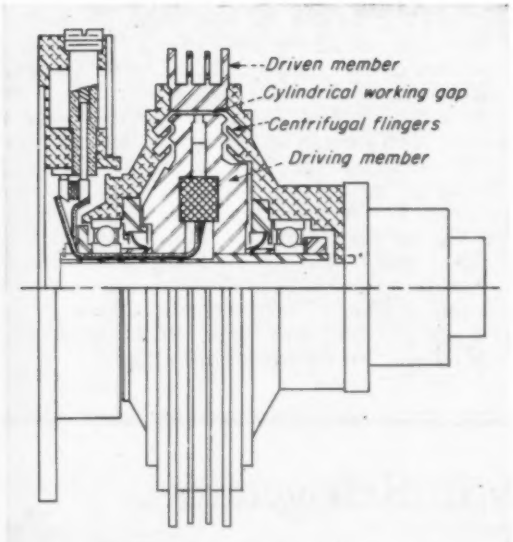
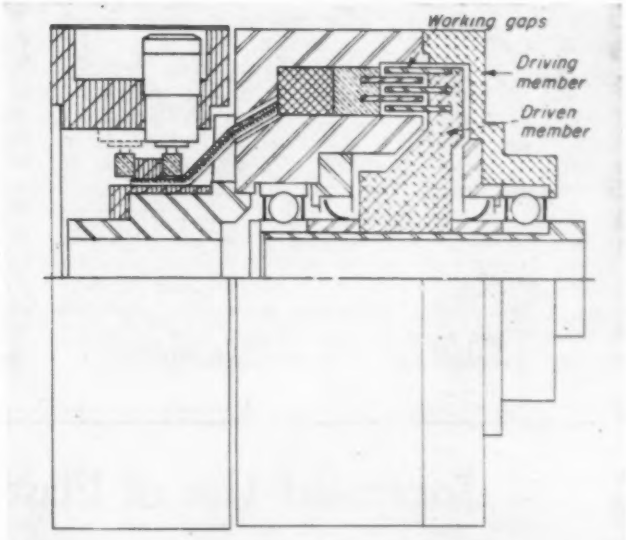


PARTIAL PART DISPLAY shows the main components of the Magneclutch. In order to dissipate the heat losses due to slip, a clutch design with fairly large surface area was necessary. This and the working gap diameter used resulted in large centrifugal forces which could pack the magnetic mixture and tie the inner and outer members together when the clutch exitation was off. In order to eliminate such packing, a cylindrical working gap and an amount of mixture slightly less than the gap volume were used. Stationary joints are sealed with rubber "O" rings.

Control Higher Power Loads



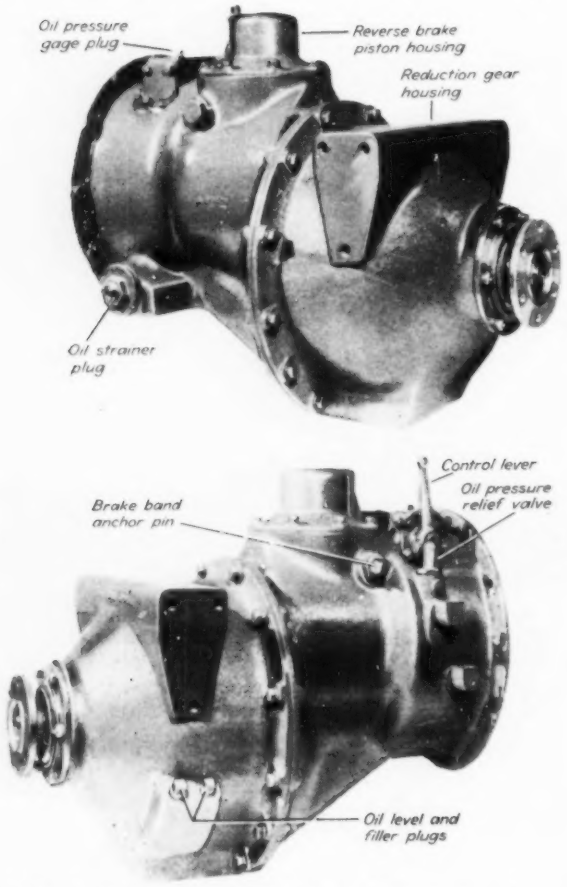
THIS DESIGN, a magnetic-particle clutch without slip rings, is used with a 2½ kw motor-generator set for constant frequency applications. Magnetic clutches do not chatter and grab on engagement, and do not have to slip, like an eddy-current device, to produce torque. To give an idea of relative clutch size, this particular design measures approximately 5½ by 9 in., overall. Other types have comparable sizes.



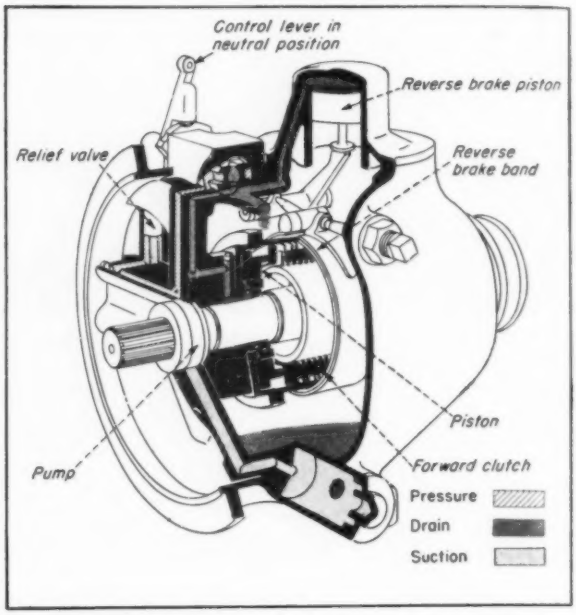
A HIGH TORQUE MODEL, this clutch is used for power take-off from a truck engine. It has an on-off duty at slow speed. There can be endless variations in clutch parts. Gaps can be cylindrical or radial. They can be single or multiple. The exciting coil can be an outer or inner member, and can be on either side, or can even be removed from either rotating member and the flux fed across an extra pair of air gaps. This last mentioned arrangement eliminates the current-collecting slip rings.

Planetary Gearing

Designed for use with 100-150 hp Packard marine engines, this "Finger-Tip Control" marine reverse gear uses a hydraulically controlled planetary gear system to



LEFT AND RIGHT VIEWS of this marine engine reverse gear show all external parts. In both views a 2.52 to 1 reduction gear assembly is also shown. These reduction gear sets are contained in a separate housing, and are not standard equipment. The reverse gear is usually installed alone, to give a direct, 1 to 1 ratio. The Packard marine reverse gear consists principally of a drive line made up by the main driving shaft and propeller driving shaft. Splined to the main driving shaft is a rotor type pump which furnishes oil for both the control and lubricating systems. Mounted behind the pump is a drum or carrier housing which contains the clutch assembly and planetary gears. A reverse band surrounding this carrier is operated by the hydraulic piston through a simple lever. There are no complicated linkages to cause maintenance trouble.



WITH CONTROL LEVER IN NEUTRAL, oil distribution to the clutch and reverse brake passages is prevented by the position of the cylindrical control valve. Channels in the valve vent both the clutch and brake passages to the inside of the housing. Since the clutch operating piston is not subjected to oil pressure the clutch plates are kept separated by six release springs. This permits the alternately spaced bronze driving disks and steel driven plates to rotate independently. The reverse brake piston is not subjected to oil pressure and the piston is held in its upward position by the piston spring. The self-expanding brake band is then in the released position and the clutch and bevel gear carrier is free to rotate. Power transmission through the rear bevel gear prevented since the pinions are not turning the rear bevel gear, but are rotating around it.

Increased Use of Plastics in Refrigerator

THE NEW CROSLEY REFRIGERATOR uses a total of 24 lbs. of plastic in 68 different parts. In a recent talk to the SPI a company spokesman, Mr. O. E. Norberg, explained where and why the parts were used.

The largest plastic part is a phenolic door liner weighing about 6 lbs. It functions as a basic structural part of the door assembly and at the same time has superior thermal properties that retard the entry of heat into the cabinet without the use of separate thermal breaker strips. It also serves as the basic structural member and helps to provide two extra feet of storage space in the door.

Plastics are also used to replace the metal shelves. Appearance is the main advantage in the use of plastics here. Crosley designers wanted to achieve a sturdy, streamlined, integrated look, and they picked a plastic to fill this need. The shelf rails are also injection molded polystyrene. The plastic rails were designed to avoid dirt-catching cracks and crevices. Also,

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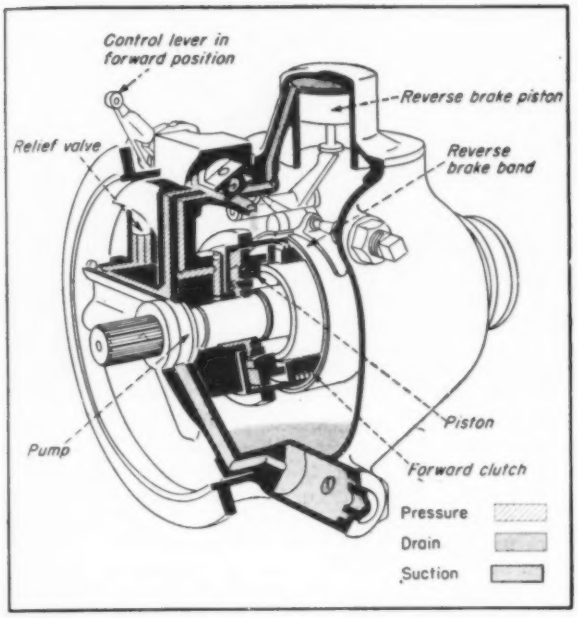
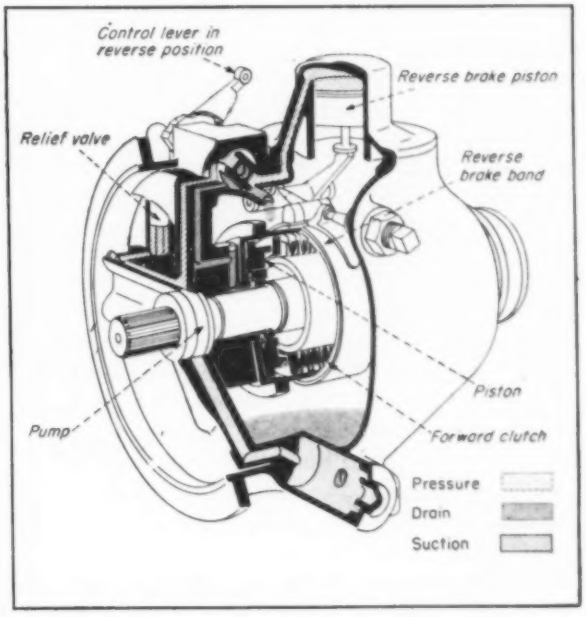
Simplifies Marine Reverse Gear

give 100 percent reverse torque. A simple control lever with forward, neutral, and reverse positions eliminates all complicated shifting. The standard reverse gear has a direct, 1 to 1 ratio. However, for boats that need a reduction in speed, Packard also offers reduction gear units with 1.45 to 1, 2.04 to 1 or 2.22 to 1 ratios.



WITH CONTROL LEVER IN FORWARD, one of the pressure channels of the control valve aligns with a drilled passage leading to the clutch operating cylinder. At the same time the reverse brake passage continues to be vented to the inside of the housing and the reverse brake remains in its released position. Oil pressure forces the clutch operating piston rearward to engage the forward clutch. Engine power transmission is then through the reverse gear main driving shaft to the clutch hub and the front bevel gear. The carrier rotates with the clutch hub due to the locking action between the driving disks and the driven plates. The clutch hub, the carrier, and the front bevel gear turn as a single unit driving the rear bevel gear and the propeller shaft by means of the trunnion and pinions. The pinions do not turn on their trunnion arms.



WITH CONTROL LEVER IN REVERSE, a pressure channel of the control valve aligns with a drilled passage leading to the reverse brake band operating cylinder. At the same time another portion of the valve vents the forward clutch oil passage to the inside of the reverse gear housing and the clutch is held in the disengaged position by the clutch release springs. The reverse brake piston, under oil pressure, pushes the brake lever down to tighten the brake band. The brake then holds the clutch and bevel gear carrier housing stationary. Engine power is transmitted through the reverse gear main driving shaft to the clutch hub and the front bevel gear. The clutch, being in released position, is free to spin within the carrier housing. The drive through the front bevel gear which rotates the pinions causes the rear bevel gear to rotate in reverse.

since the color is built in, there is no danger of canned goods or bottles wearing out the finish.

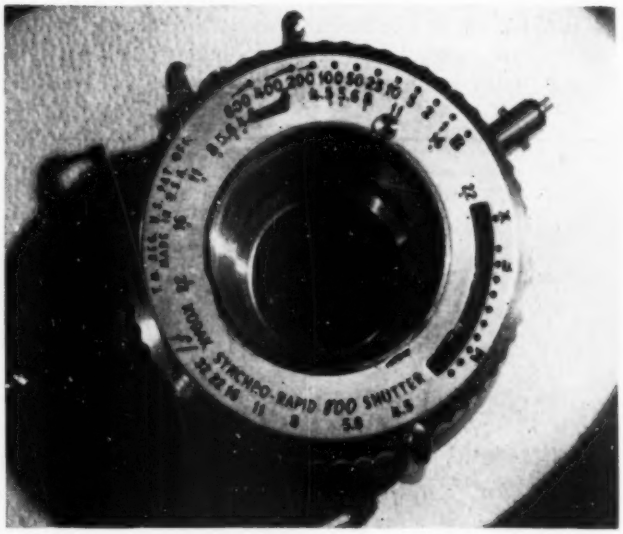
A butter safe, which keeps butter at just the right temperature for easy spreading, has a body, door, and tray injection molded from heat resistant polystyrene. Here, the insulating properties of plastics are used in addition to a controlled heating element to maintain proper temperatures.

The freezer door is made of two injection molded polystyrene halves cemented with Fiberglas insulation between them. This is an ideal plastic application, since the structural parts are also insulators. They help to prevent condensation of moisture and subsequent icing more effectively than metal doors. Transparent plastic is used to make meat keepers and crispers because an opaque material would destroy the impression of spaciousness that Crosley designers wanted to convey. They also chose a clear plastic in this case to give an impression of sanitation and cleanliness without a hospital feeling of sterility.

Among other major uses, Crosley relied heavily on plastic for decorative effects, especially with gold and silver plated acrylic inserts. A Nylon roller in the latch mechanism was selected for its wear properties. The new refrigerator is built by Crosley Div. of Avco Mfg. Corp., Cincinnati, Ohio.

Redesigned Camera Shutter Has Double

Photographic shutters have traditionally been divided into three main groups: The "flip-flop", or box camera type; the "Between-the-lens" type, which is the most popular; and the "focal plane" shutter, which is a slotted curtain or variable slit type shutter. The focal plane shutter is by far the fastest of the three types, but is not widely used. It has been the dream of shutter designers for years to make a between-the-lens shutter with a better range of shutter speeds—while still retaining the many characteristics that make this shutter ideal for most cameras. The new Kodak Synchro-Rapid 800 shutter, which is now featured on a new model of the Kodak Tourist Camera, is the realization of this aim. It has a top speed of 1/800 sec., and is built on an entirely different design principle. For its size, it is twice as fast as the fastest between-the-lens shutter previously made. Its top speed is within 20 percent of the top speed of most focal plane shutters. In addition to the new shutter design, the Kodak Synchro-Rapid 800 Shutter also features a front and back lens element mounted in a machined tubular section joined to insure accurate lens alignment. At the present time this new shutter design is available in only one size. Other sizes will follow.



REDESIGNED KODAK Synchro-Rapid 800 shutter looks much the same as before, but has twice the usual shutter speed. In addition to the new speed the unit also has these new features: A uniform trigger stroke, with or without the flash attachment, which is advantageous when limited-power solenoid tripping devices are used; a continuous time delay adjustment aids operation with a flash lamp at any shutter speed; the shutter flash contacts will stand a load of 15 amps at 25 v—this is important in modern practice when multiple flash bulbs are used.

New Model Coffee-Maker Has

AFTER REDESIGN: The "Tricolator" not only has better appearance, but costs less to manufacture and sell. Cushing & Nevell, an industrial design firm in New York, redesigned the coffee-maker for the Tricolator Corp., also of New York City. The new utensil now stands in New York's Museum of Modern Art as a part of a permanent industrial collection. New Tricolator has only five major parts, requires only one self-tapping screw, and only one assembly operation. This is in contrast to the old design, which has twelve major parts, eight rivets, and required ten assembly operations. The top section is spun from aluminum both to give a better appearance and as a manufacturing economy.

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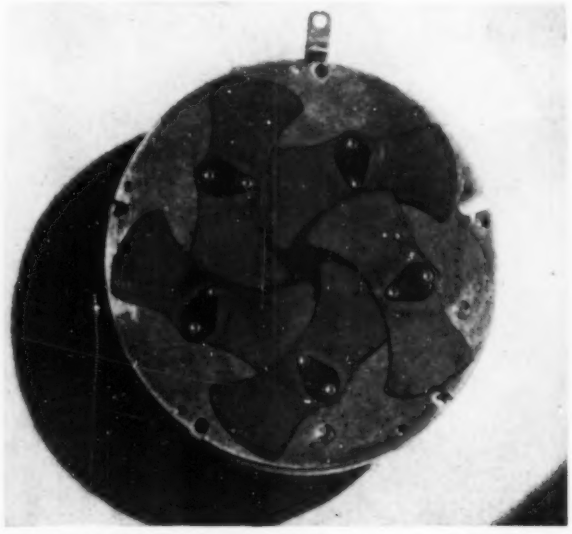
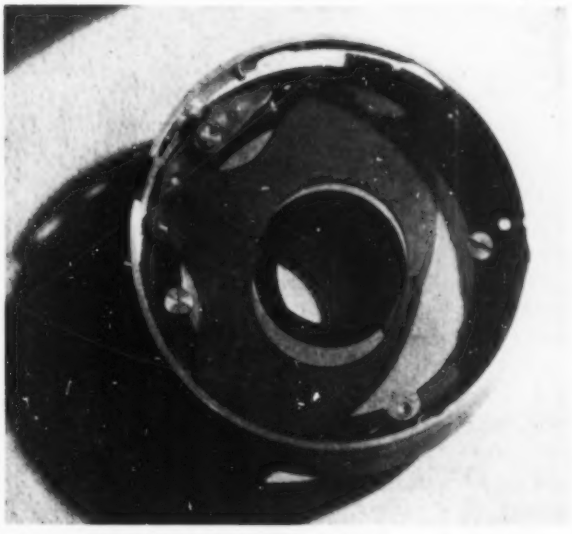
Conventional Speed, Longer Life



THE MAJOR DIFFERENCE between the conventional and the new shutter design is in the motion of these blades. In a conventional shutter, blades move first in one direction, to open, and then in reverse, to close. This change in direction of motion limited operating speed and required use of extra-heavy spring. In the new design each blade moves continuously in a counter-clockwise partial circle. The rear part of each blade swings up, to open the aperture, and the fore part swings down to close it again.



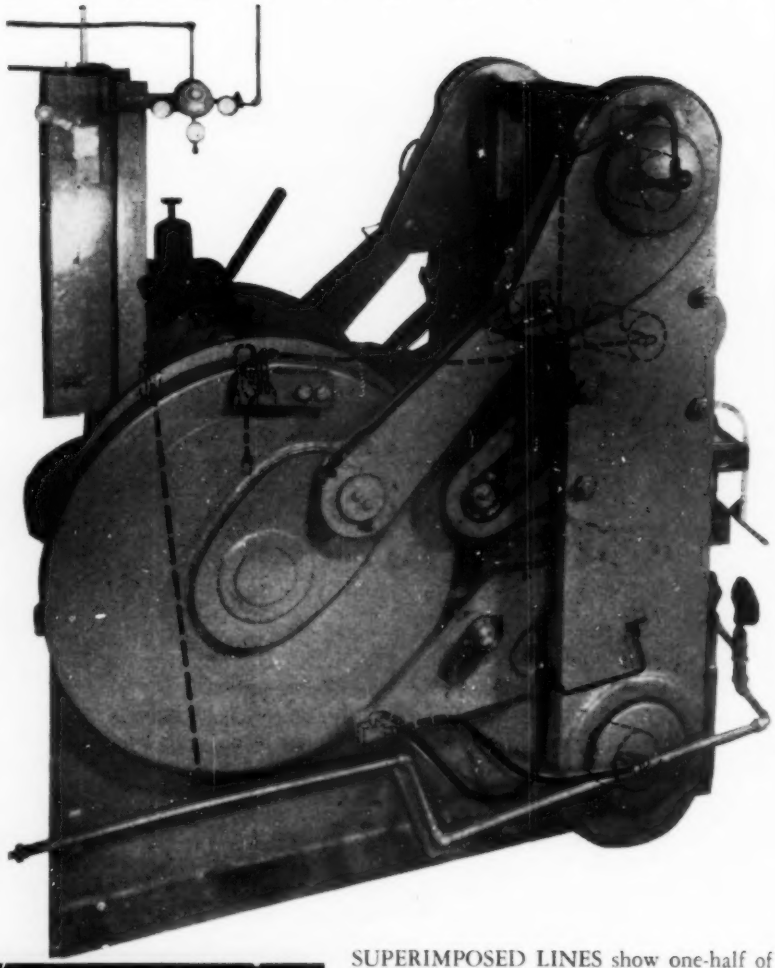
SINCE THE APERTURE MUST OPEN while the shutter is being cocked, a pair of cover blades (above) are used to shield the lens during the setting or cocking cycle. These cover blades are located behind the shutter blades. Initial pressure on the shutter release lever moves them out of the way before the shutter is tripped. These cover blades also act as an extra precaution against light leakage. This is important with present day very fast film, which is extremely sensitive to small amounts of light.

Improved Appearance, Fewer Parts

BEFORE REDESIGN: The Tricolator was recognized as a reliable but unwieldly utensil. Both the new and old coffee-makers use filter paper to strain the sediment from coffee. Both also have the feature that ceramic parts can be used alone as a coffee or tea-pot. The company reports that one month's sales increase paid the cost of redesign. Before redesign the Tricolator had twelve major parts, eight rivets, and required ten assembly operations. It was priced about 50 pecent higher than the new design. The coffee maker was originally made in 13 colors. This number has now been reduced to 7, which means a substantial saving in manufacturing and inventory cost.

New Grease System for Goods Press

The Model 800-32 Single Mechanical Goods Press, made by The McNeil Machine & Engineering Co., of Akron, Ohio, is used for the curing of mechanical rubber goods such as automobile trim, motor mounts, heels, soles, mats, and other such parts. Loads as high as 800,000 lb. are developed between platens, and this means a high unit bearing load. Also, curing time for various good can vary from 3 to 40 minutes, resulting in a variety of lubricating frequencies. Heretofore it was necessary to manually grease 15 bearings; always with the possibility of overlooking an important bearing. Now, an automatic system measures the correct amount of lubricant for each bearing and assures complete lubrication during each cycle.



SUPERIMPOSED LINES show one-half of the lubrication system. Tubing goes from the master cylinder, through the master lines, to the individual manifold injectors, and from the injectors to the individual bearing. Each manifold is individually adjustable as to the volume of grease dispensed to each bearing. With the new setup, the master cylinder builds up to its maximum pressure and discharges all injectors at every other operation of the press. After all the manifolds have been properly adjusted, the only maintenance requirement is the manual refilling of the master grease cylinder. Lincoln Engineering Co., of St. Louis, Mo., furnishes the centralized lubrication components used on this machine.

THE MASTER PUMP UNIT is actuated by the movement of the top platen support of the press at each opening and closing. The small roller at the bottom of the actuating lever is engaged at all times in a surface contact with the top platen support so that the vertical movement of this support actuates the master cylinder pump. The pump discharges at the end of two complete cycles of the press.

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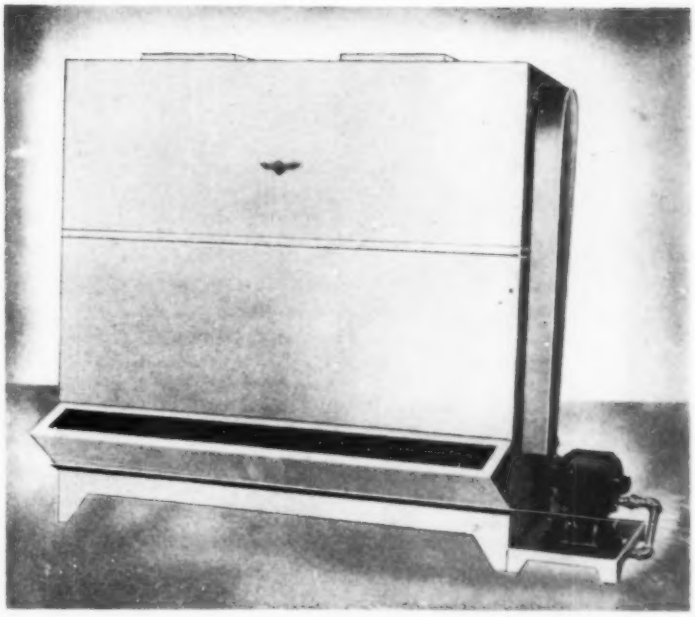
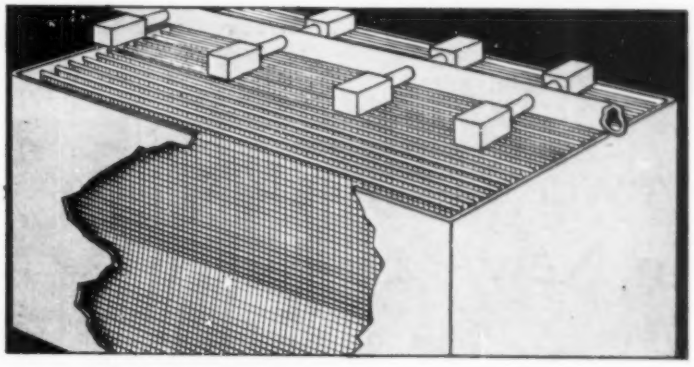
Cooling Tower Cuts Waste

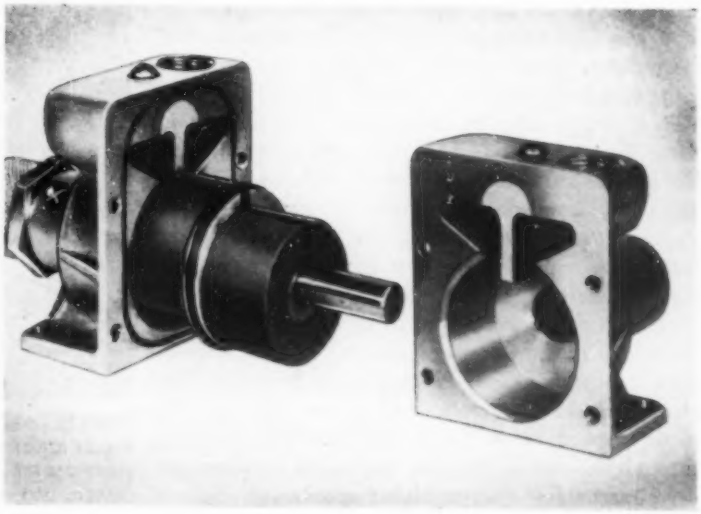


The "Hydro-Miser", a new cooling tower made by Larkin Coil Co., of Atlanta, Ga. uses a new type of wetted surface to get sufficient cooling with a low water consumption. The evaporative condenser has a grid-work of electro tinplated bronze wire screens, with each screen set at a slight angle. Water is sprayed on these screens and is then evaporated and cooled by a blast of air drawn in by fans. The air flows on both sides of each screen in the grid-work, to give twice the usual cooling surface. The grid-work is housed in a hot-dip galvanized steel case and the entire unit can be removed for cleaning. Other features claimed are: a minimum concentration of salts and deposits in the sump tank; hot-dip galvanized sump tanks, fan scrolls, wheels, frames, and panels; and large orifice-type spray nozzles.

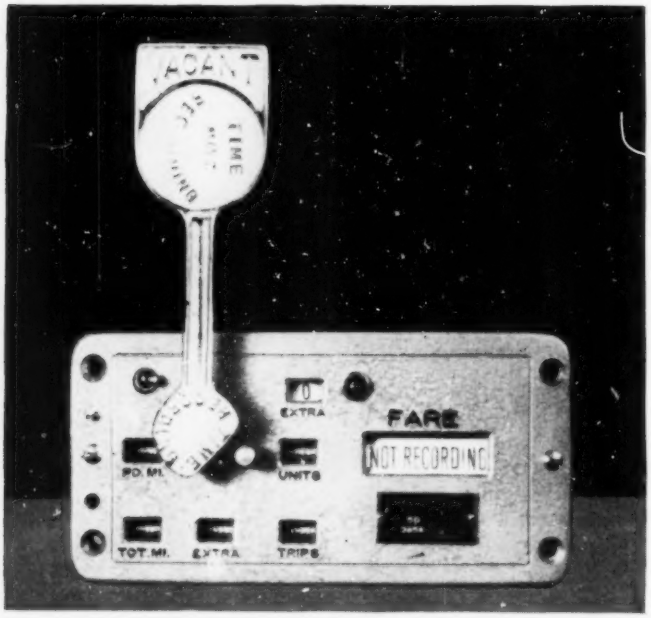


CUT-AWAY drawing shows the grid-work of electro-tin-plated bronze wire screens used in cooling. Water is sprayed under pressure onto the screen and clings to the fine mesh of the wire. Centrifugal-type fans pull air in from the bottom of the tower. The air flows on both sides of each screen and evaporates the water.

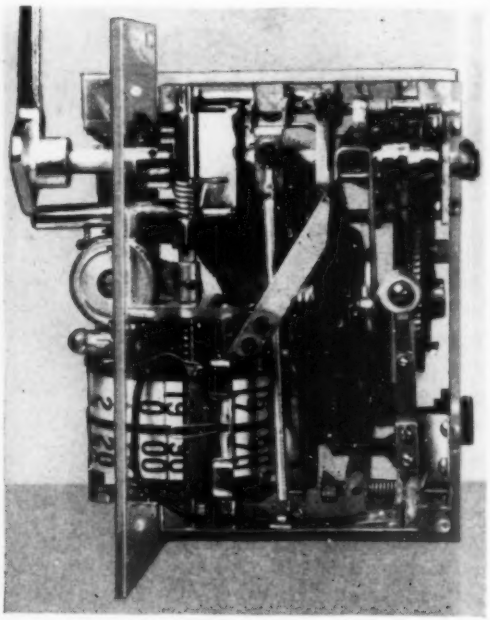
Nylon Piston Resists Abrasion



In designing a multi-purpose pump, the ECO Engineering Co., Newark, N. J., needed an eccentric piston that was light, abrasion resistant, that would maintain high pressure and volume, work quietly without lubrication, and have a high degree of chemical resistance. Pistons machined from molded nylon met these specifications, and are now being used. The DuPont nylon rod was made by the Polymer Corp., of Reading, Pa.



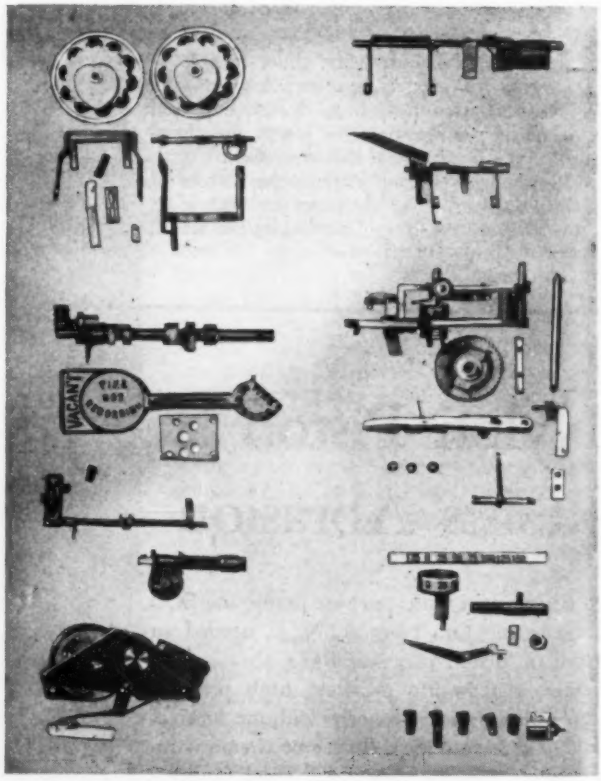
NEW "CABOMETER" comes in five models to meet the requirements of different taxi operations. Lightweight and mechanical simplicity are the main design features of the new meter. This outside view shows the seven recording dials normally needed.



VIEW OF THE NEW METER in a transparent case gives some idea of the compact, simplified construction. The new unit has more than a hundred fewer parts than conventional meters.

Fewer Moving Parts In New Taxi Meter

Design simplicity and easy maintenance were the objectives when Cabometer, Inc., of Anniston, Ala., built this new taxi meter. The company claims a reduction of over 100 moving parts in the new meter compared to standard designs. Since meters are usually mounted in the dash of the taxi, lightweight alumium construction was used wherever possible. The meter mechanism differs from traditional designs in that the principle operating member is a screw-type cam. A number of pins fit into the cam grooves, and as the cam turns the movement of these pins register the distance the cab has traveled. Motive power is provided either from the transmission or from an adaptor mounted on the speedometer. At the same time that the indicator wheels are tripped to register the fare for the customer visually, a gear arrangement turns another dial to make a permanent record of the distance traveled. Since waiting time is also a factor in cab operation, the Cabometer is fitted with either a springwound or electric clock. The clock mechanisms operates the meter at speeds below 8 mph, with the clock operating through the cam in the same way as the speedometer drive mechanism does at higher speeds.



IMPORTANT PARTS of the new meter are displayed above. Gears are case hardened, and cams, levers, and other moving parts are of brass. Most stamping are of cold rolled steel. Where necessary, all internal parts are cadium plated.

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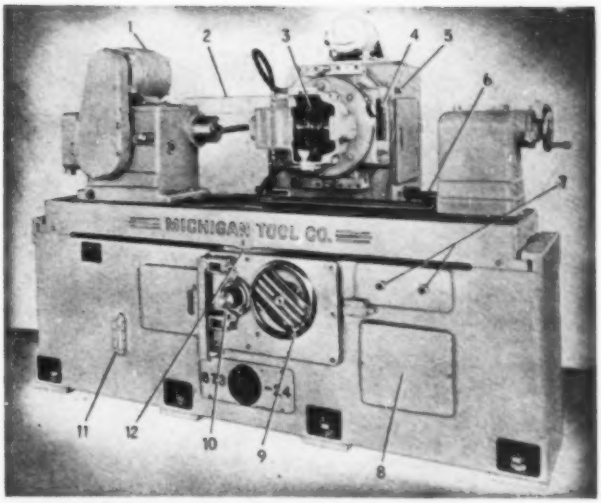
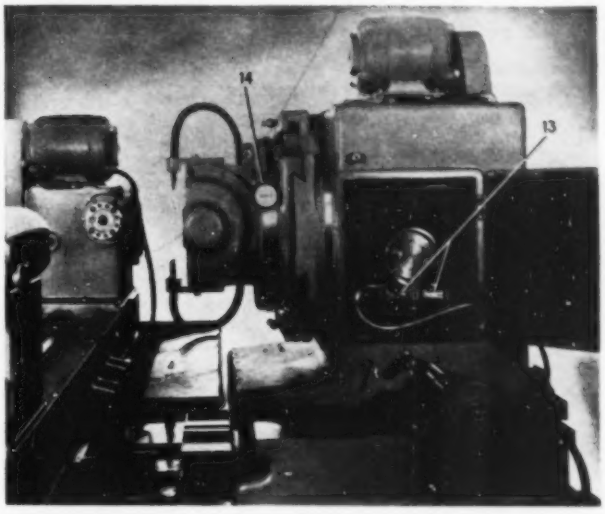
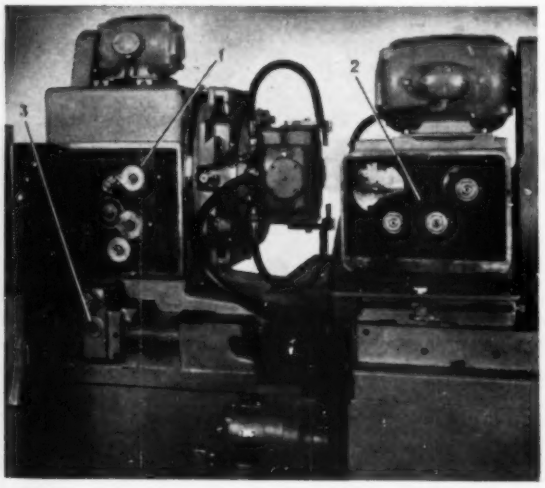
Gear Shaver Is Fully Automatic

Made especially for production work, the model 873 Gear Finisher will shave spur and helical gears and involute splines by either the Underpass, the Traverse, or the Traverpass methods. The shaving method depends on the size and type of gear to be finished. After the machine has been set up to finish a particular gear the operation is completely automatic. An unskilled operator has only to load and unload the machine and push "stop" and "start" buttons. The machine will curve-shave (crown) wide face gears as well as narrow ones. Machine control is entirely electrical, which simplifies operation and increases flexibility and precision of control. The cutter head movement is controlled by limit switches. Another feature is that the machine is so designed that an overhead crane or hoist can be used to load heavy gears. The new model 873 Gear Finisher is made by Michigan Tool Co., of Detroit, Mich.

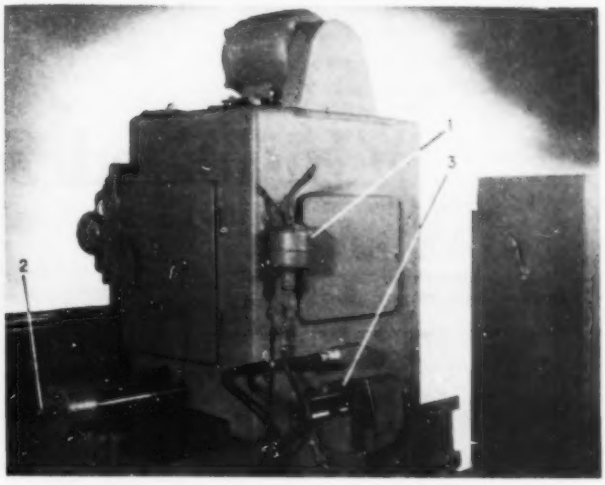


GEAR FINISHER PARTS are: (1) spindle drive motor, (2) electrical control panel, (3) shaving cutter, (4) cutter reciprocation slide, (5) cutter slide angle positioner, (6) head slide, (7) start and stop buttons, (8) infeed motor and drive, (9) handwheel for setting center distance, (10) stroke limit adjustment and limit switches, (11) cutting oil gage, (12) pivot for transverse curve shaving, (13) limit switches for cutter reciprocation, and (14) dial indicator for crossed axis setting.





ROTATION SPEED and speed of cutter reciprocation for any of the three methods are regulated by change gears, (1 and 2). Take-off point (3) is for special curve-shaping attachment for transverse shaved gears.



SLIDES AND WAYS are lubricated by a one-shot lubricator, (1) Manual control, (2) is for setting head position on new setup. Limit switches on back of machine, (3) are for transverse shaving. They are actuated by trip dogs.

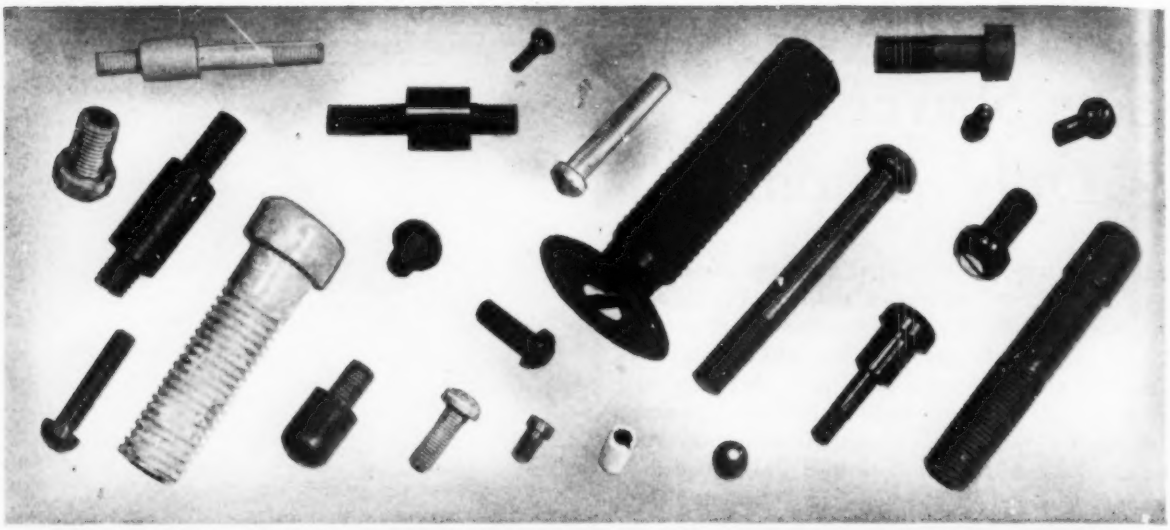


Fig 1—Insulated screws, with a metal core and a plastic exterior, are shown in a variety of special and standard types. While standard shapes and sizes will suit most applications, an economy-minded designer can often specify a special insulated screw that will serve several functions in addition to the main fastening requirements.

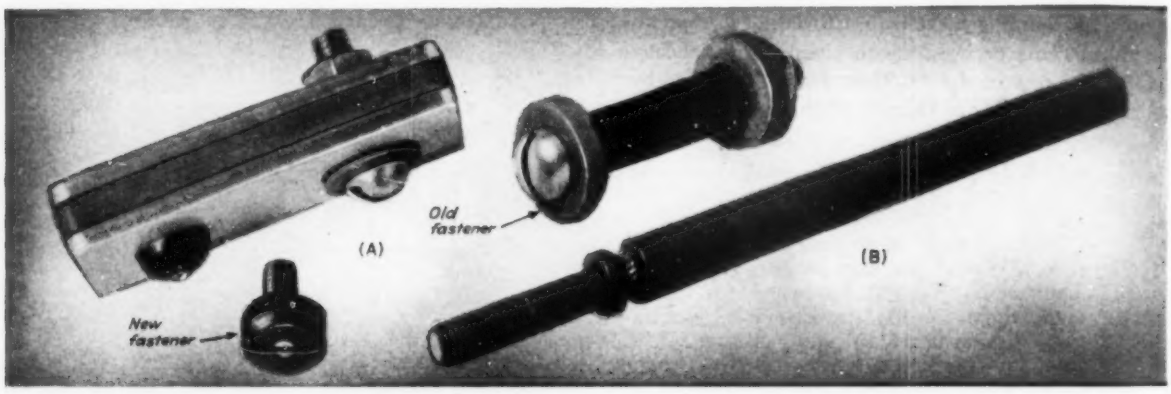


Fig. 2—(A) Five parts are needed to provide electrical insulation by ordinary fastening methods. The one insulated screw replaces two washers, an insulating boot, and a steel nut. (B) This composite metal-plastic rod has one screw already machined. Next step is to cut off screw and slot the head. Notice the knurled metal core.

Insulated Screw Fasteners for

B. G. FORMAN

Research Director
Forman Insulating Screw Corporation

IN MANY ASSEMBLIES, the type of fastener that is used to join parts and components to the main frame places severe restriction on the designer. Fasteners for electrical or electronic assemblies perhaps are the best illustrations for the case in point: Joining two metallic parts with a metallic screw or bolt brings the two parts into direct electrical contact, often when such contact is not desired. Or the use of metallic fasteners for joining two metals with widely different activities in the electrolytic series may cause electrolytic corrosion and resultant failure of the assembly in service.

The usual solution to such problems is to insulate the components from each other by a strip or panel of nonconductive material, and to use insulated bushings, washers, boots, or other devices to prevent electrical conductivity. Another solution is to use a plastic insulated fastener, as shown in Fig. 1, that in itself is non-conductive. A composite metal-plastic screw such as this automatically insulates the various components, and therefore does away with the need for auxiliary insulating devices. While not so strong as a screw made of one of the higher tensile strength allows, insulated fasteners are suitable for many jobs in which a moderately high holding strength is sufficient.

Other advantages of plastic insulated fasteners are: (1) being made on screw machines, threads are compara-

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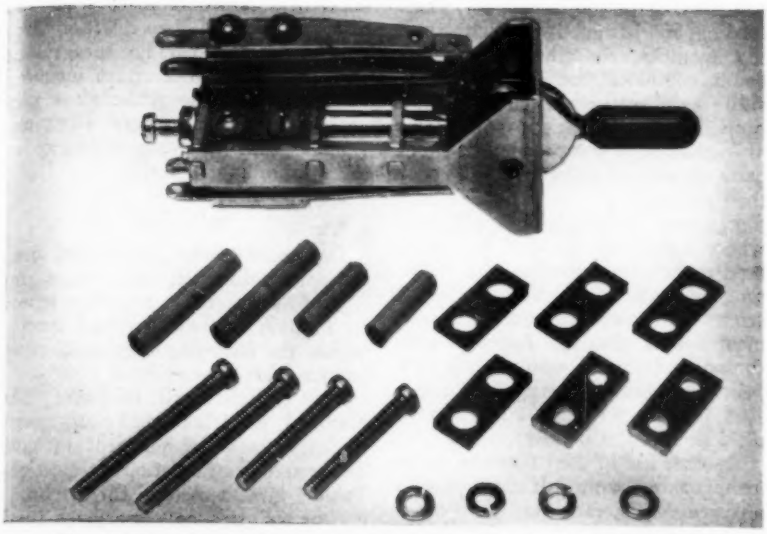

Fig. 3—Four insulated screws replaced 18 parts in this anti-capacity switch. A further product simplification would result if the component were redesigned to take full advantage of the insulated screws.

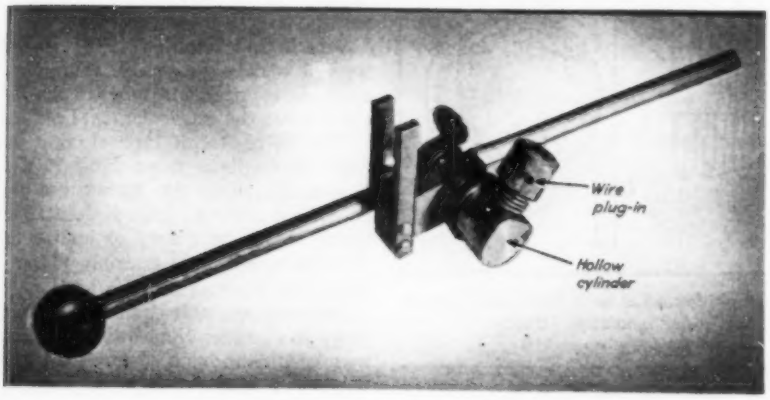

Fig. 4—A composite screw is used as a gravity switch. A metal ball in a hollow cylinder contacts the screw core to complete a circuit. A wire lead-in connects to the metal core through a hole in the plastic head.

Electrical Equipment

ble in accuracy to those cut in metal; (2) the fasteners are highly resistant to damage by shock and vibration; (3) they are resistant to many chemicals since the plastic can be specifically selected for resistance to certain chemicals; and (4) they are light in weight and can be made in colors to match or complement the basic structure.

Construction and Application

Basically, a composite metal-plastic screw consists of a serrated metal core that has been extrusion coated with a plastic exterior to form a composite rod. The rod is then cut and machined on an automatic screw machine to form a precision screw having any desired number of threads, head shape, length, or color. A composite rod is shown, Fig. 2, with one screw already formed. The metal core gives the screw a safe-driving, unbreakable head, while the plastic exterior provides insulation.

Wall thickness and type of plastic determine the dielectric strength of the insulation. Any extrudable plastic can be used, as well as any type of metallic core. Screws using eight well-known plastics have been developed. Of these eight, four are now commercially available as stock products, and will meet the majority of applications. They are cellulose acetate, polyethelyne, cellulose acetate butyrate, and ethel cellulose. Each group has definite properties that fits it for specific classes of application.

Fields of application of these fasteners are broken into three main groups: (1) Electrical and electronic components (2) Chemical equipment and (3) Mechanical Components.

In the electrical field, applications are many. Any one point on the screw surface is fully insulated from any other point, allowing the screw to be used as an insulator, separator or feed-through insulator as desired. In many applications of this type, the composite screw may be made with special cores allowing for non-magnetic or special magnetic qualities, as desired.

In an anti-capacity switch, Fig. 3, four composite screws replace eighteen formerly used parts. If the component were re-designed to take full advantage of the insulated screws, a complete new design approach would be possible, and a more compact, simple component would result.

An electric stop-motion device, Fig. 4, shows the non-conductive fastener in a multiple role. The composite screw is part of a gravity switch. A ball inside the cylinder contacts the metal central portion of the composite screw and thereby sets off the electrical stop-motion device. The plastic portion of the screw insulates this assembly from the rest of the circuit. The non-conductive fastener is used as an adjustment to control the length of travel of the ball within the cylinder. This application shows three functions of the screw: (1) The metal core is used as a contact point. (2) The plastic exterior insulates the switch from the rest of the assembly. (3) The composite screw is used as an adjustment device. The fineness or sensitivity of the adjustment may be varied by the use of different threads.

In another electrical application a composite screw one-inch in diameter and twenty-four inches long is used as a lead screw for a metal-detecting machine. In another case the screw replaces the threaded insulating core in a television inductance coil. In this application the use of a proper core allows a fine permeability tuner to be built with a minimum of parts. Electrical uses run from simple jacks and relays through and including such complex assemblies as the electronic sequence calculators.

For applications in the chemical and food industries, the choice of

plastics and metals available makes possible the production of chemically resistant fasteners. The plastic exterior of these composite screws can be used as a liquid seal in containers. The threads of the screw act as a multiple unit of washers and in effect, produce a seal. This application is significant where volatile chemicals are used. In the field of refrigeration, for example, the composite screw can be used to replace the insulating bushings and extruded washers that prevent loss of refrigeration and chemical refrigerants. Furthermore, when used in or around food containers they import no odor, stain, or taste and can be made fully resistant to food acids.

In the mechanical design field the ability of the composite screw to stand up under shock loading and vibration are important features. In general, a soft flowing plastic is best for shock loading, and has good vibration damping qualities. Polyethelene, for example, combines good shock resistance with excellent electrical insulation properties. Where these two problems are combined in one product the slight extra expense of polyethelene is more than justified.

In one mechanical application, where a metal screw and nut were used as the adjustment for each stitch-cam on a knitting machine, the metal screws were constantly out of adjustment due to vibration. Redesign with composite fasteners, using a soft flow plastic to absorb the vibration and a steel core to hold the stitch-cam in place, solved this problem.

Another use of the composite screw manufactured with soft flow plastics occurred in the mounting of components of a recording machine. Sound distortion caused by vibration was eliminated. When radio speakers or other audio reproducers are mounted with the new fastener, the screw prevents the low frequency vibrations from causing annoying resonance in the surrounding cabinet.

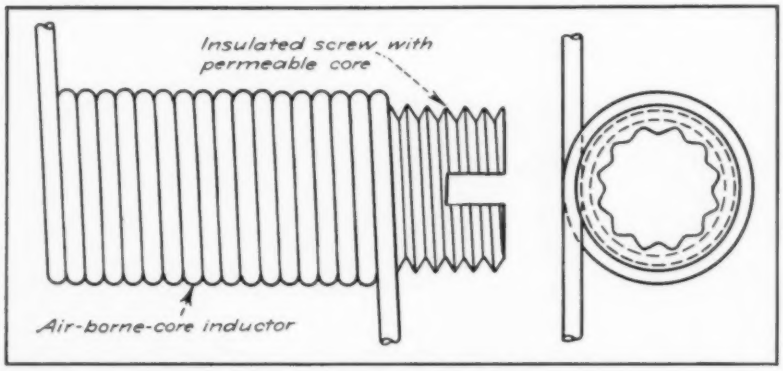

Fig. 5—Using a permeable core, this insulated screw forms the basis for a television inductance coil, which can easily be incorporated into a fine permeability tuner.

Table I—Plastic Wall Thickness With Various Core Sizes

Screw Sizes	Core Sizes, in.												
	5/64	3/32	7/64	1/8	9/64	5/32	11/64	3/16	13/64	7/32	15/64	1/4	17/64	9/32
8–32	0.022													
10–24	0.029	0.021												
10–28	0.033	0.025												
10–32	0.036	0.028	0.020											
12–24	0.042	0.034	0.026			Plastic Wall Thickness								
12–28	0.046	0.038	0.030	0.022										
1/4–20	0.053	0.046	0.038	0.030	0.022									
1/4–24		0.051	0.043	0.035	0.028	0.020								
1/4–28		0.055	0.047	0.039	0.032	0.024								
5/16–18				0.058	0.032	0.042	0.034	0.026						
5/16–20				0.062	0.050	0.046	0.038	0.030	0.022					
5/16–24				0.067	0.054	0.051	0.043	0.035	0.028	0.020				
3/8/16					0.059	0.068	0.061	0.053	0.045	0.038	0.030	0.022		
3/8–20							0.067	0.060	0.053	0.046	0.038	0.030	0.022	
3/8–24								0.067	0.059	0.051	0.043	0.035	0.028	0.020

Screw Sizes	Core Sizes, in.												
	5/32	3/16	7/32	1/4	9/32	5/16	11/32	3/8	13/32	7/16	15/32	1/2	17/32
7/16–14	0.096	0.080	0.064	0.048	0.032								
7/16–20		0.094	0.078	0.062	0.046	0.030							
7/16–24		0.099	0.083	0.067	0.051	0.035		Plastic Wall Thickness					
1/2–13			0.092	0.076	0.060	0.044	0.028						
1/2–20				0.092	0.077	0.061	0.045	0.030					
1/2–24				0.099	0.083	0.067	0.051	0.035					
9/16–12				0.103	0.087	0.071	0.055	0.040	0.024				
9/16–18					0.106	0.090	0.074	0.058	0.042	0.026			
9/16–27						0.102	0.086	0.070	0.054	0.038			
5/8–11						0.099	0.083	0.067	0.051	0.035			
5/8–18							0.106	0.090	0.074	0.058	0.042		
5/8–27								0.102	0.086	0.070	0.054	0.038	0.022

Selection

Composite screws are selected first, by length, diameter, head shape and type of thread, just as any other screw. In addition to these standard specifications the following special selection factors must be considered:

First, plastic wall thickness. For electrical insulation the plastic wall thickness must be sufficient to give the required insulation protection. Table I gives a method for determining the wall thickness of a given sized screw, with various choices of cores. For a complete discussion on the thickness of plastic insulating materials see PRODUCT ENGINEERING, April 1950, pages 135-139, "Laminated Plastics Insulation" by P. E. Littlefield.

The second selection factor is the type of metal core (steel, aluminum, brass, or copper) and type of plastics needed for the particular application.

The third factor in selection is color. Since there is no limit to the choice of colors, the screws can be chosen to give a desired decorative effect, or can be chosen as an aid to color identification in assembly of complex parts.

Presently available composite screws range in diameter from No. 8 to $\frac{1}{4}$ in. with cellulose acetate; from No. 10 to $\frac{3}{8}$ in. with a polyethelene insulation; from No. 8 to $\frac{3}{8}$ in. with cellulose acetate butyrate; and from No. 8 to $\frac{1}{2}$ in. with ethel cellulose insulation.

The following applications of the composite screw give examples of the varied properties available:

Sulphuric acid corrosion, light load, high voltage—Polystyrene.

Oil seal in transformer, no load, some vibration—Cellulose Acetate Butyrate.

High impact, high voltage—Ethel Cellulose.

High voltage, high vibration (excursion 0.050)—Polyethylene.

Heat 140 F, moderate voltage, moderate load—Cellulose Acetate.

High load, high voltage, moderate vibration—Ethel Cellulose.

Atomizer seal, wear resistant—Nylon. Vinyl chloride acetate.

Cigarette lighter seal, wear resistant—Ethel Cellulose.

Glass connectors, vibration, high load—Ethel Cellulose.

Heat conductivity, light load—Polystyrene.

SYNTHETIC HYDRAULIC FLUIDS

Possibilities of new types of hydraulic fluids are compared with modified petroleum oils, for aircraft and industrial applications. Flammability is of primary concern, but relative viscosity, effect on metals, toxicity, operation at high and low temperatures, volatility, and effect on seals and packings are also considered.

C. M. MURPHY* and W. A. ZISMAN*

Naval Research Laboratory

FLAMMABILITY OF LUBRICATING and hydraulic fluids is frequently a hazard in aircraft and industrial equipment. Because of the adoption during the last war of the lower boiling petroleum-base fluids thickened with polymers, which satisfied the demand for a fluid capable of operating over wide ranges of temperature, fire hazards are even more acute. Fires and explosions caused by hydraulic fluids were significant sources of casualties and losses of aircraft equipment. In industrial practice, too, where hot equipment or parts are handled by hydraulic machinery, such as in steel mills and die casting shops, there has been an increasing interest in preventing fire and explosion hazards caused by sudden leaks or component failures.

A new laboratory method has been developed for measuring the flammability of a fluid under empirically controlled conditions, in terms of the volume percentage of oxygen necessary in an oxygen-nitrogen mixture to permit an arc-ignited flame to propagate along a metal tube filled with a finely dispersed mist or spray of test liquid. A high V.I. (viscosity index) low pour-point, petroleum-base hydraulic oil always exploded when the oxygen concentration was 12 percent, while a highly halogenated hydrocarbon required at least 60 percent oxygen. The first fluid forms explosive mists in the air, since air contains approximately 21 percent of oxygen. The minimum oxygen concentration required to form an explosive mixture is designated as the "oxygen demand."

To verify conclusions obtained with the spray flammability apparatus, tests were made by the Navy on the ability of 50 caliber incendiary bullets to ignite each proposed fluid. This work confirmed the predictions based upon the laboratory flammability tests.

All of the five flammablity test methods—flash point, fire point, spontaneous ignition temperature, spray flammability, and incendiary fire—have been applied by the authors to the study of a wide variety of hydraulic fluids, synthetic oils, and organic liquids of promise. It is noteworthy that fluids having an oxygen demand of less than 40 to 50 percent in the spray flammability test proved explosive in the incendiary fire test, while those requiring appreciably more than 50 percent of oxygen did not explode.

From these tests it was shown that no available petroleum-base or castor-oil-base hydraulic fluid is sufficiently flame-resistant. The addition of antioxidants in high concentrations does

Table I—Properties of Modified Petroleum Hydraulic Oils

Property	Navy Department Specification 14-0-19 Grade 2075 H*	Navy Department Specification 14-0-19 Grade 2110 H*	Navy Department Specification 14-0-15 Grade 2190 T*	Bureau of Ordnance Specification 51-F-21	Bureau of Ordnance Specification 51-F-23	U. S. Army Spec. No. 2-79B	Air Force-Navy Aero. Specification AN-O-366
Flash Point, F min	315	325	350	225	300	225	200
Fire Point, F min	345**	360**	380**	235	325	240**	225**
Cloud Point, F max	0**	0**	.....	−35***	−35***	.....	−65***
Pour Point, F max	<0**	<0**	35	−40	−40	−50	−75
Viscosity, cs at 210 F min	5**	6.2**	8.5**	10	8	10	5**
130 F min	13.1–18.2	18.2–25.2	39.8–44.2	20**	22**	24**	10
100 F min	32**	48**	109**	27	40	38	14.5**
0 F max	2180	4360	.....	215	1500	600**	110**
−25 F max	.....	.....	.....	500	7500	2200**	220**
−40 F max	.....	.....	.....	.....	.....	7500	500
Viscosity Index	75	75	25**	185**	153**	170**	225**

* Oils of same viscosity grade and approximately the same characteristics, but without rust inhibition properties are also specified.
** Not a specification requirement, but typical of oils conforming to this specification.
*** 72-hour storage test.

* *The opinions expressed in this article are those of the authors and are not to be construed as official opinions of the United States Navy Dept.*

not sufficiently improve the resistance to all types of fire hazards studied.

Petroleum oil mists explode readily in the atmosphere. Conditions favoring explosions were created at pressures as low as 25psi with all commercial high V.I. hydraulic fluids in use (during the war) and with oils below the SAE 30 viscosity grade. Few liquids were resistant to all the laboratory tests for flammability, although many were resistant to one or more tests. The most serious fire hazard found was reproduced by the spray flammability apparatus and the results showed that the ignition of a mist or fog of lubricant in the atmosphere (such as could occur following a break in a high-pressure hydraulic line) can cause a serious explosion.

Few liquids in the desired range for hydraulics are truly non-flammable. Nearly all organic liquids pyrolize or oxidize if exposed long enough to temperatures over 500 F; the products formed are often flammable. In the search for completely nonflammable liquids, two other types of fluids have not been overlooked. These are: (1) Flammable under extreme conditions, but will not support combustion when the source of ignition is removed; and (2) flammable, but still so much less hazardous than available fluids, to deserve attention as a "less flammable" fluid.

Advances in Conventional Fluids

By properly refining and selecting petroleum fractions, fluids having little tendency to form gums and sludges under common operating conditions have been prepared. Additives have been developed to decrease wear, and improve oxidation stability. Rusting caused by condensation of water or leakage can be avoided by adding inhibitors, and effective additives have been found to reduce foaming. Where necessary to obtain high V. I. fluids, oil-dispersible linear polymers such as the polybutenes and the polyesters, when added in minor concentrations, increase the viscosity and improve the viscosity-temperature characteristics.

The wide variation in petroleum-base hydraulic oils is shown in Table I, which gives the requirements of some military specifications. Grades 2075 H, 2110 H, and 2190 T of Table I are rust inhibited petroleum oils that generally contain an oxidation inhibitor and are commonly used for applications where mild temperature conditions prevail. The lightest viscosity grade 2075 H can be used under winter conditions. Comparable viscosity grades without the rust inhibitor are used where danger of water contamination is unlikely. The 51-F-21 oil is a low temperature, polymer thickened, petroleum-base fluid with a high V.I. (small temperature coefficient of viscosity). It is required in applications where the hydraulic volumetric efficiency would otherwise be decreased at low temperatures if straight petroleum base oils were used. The 51-F-23 fluid has even a higher V.I. and is useful in systems which are particularly sensitive to changes in volumetric efficiency. The other two fluids shown in this table are also polymer thickened petroleum fluids which are required for the wide temperature range over which they are used. The latter of the two (AN-0-366) is an aircraft hydraulic fluid which can be used at temperatures below —40 F. Its range of operation is, however, limited to a maximum of 200 F, since a low viscosity, volatile-base oil is required for the desired low temperature properties.

Although the hydraulic fluids of Table I are military specification oils, similar fluids are available from the oil companies under their own trade names for commercial applications.

Diester Type Fluids

Aliphatic diesters, developed by the Naval Research Laboratory during the last war are low volatility, low pour point, high V.I. synthetic oils for use in liquid and grease lubricant compositions. They are widely used for instrumentation, precision equipment and aviation control bearings operating over the temperature range —80 to 250 F. Many of the diesters in the low viscosity, low pour-point ranges are much less volatile and have higher flash points, fire points and S.I.T.'s than comparable petroleum oils. Despite their comparatively high flash points, the oxygen demand differs little from that of petroleum oils of comparable viscosities, so they are quite flammable under some circumstances.

They are equally responsive to the same types of anti-oxidants as mineral oils and the resulting solutions are just as stable to atmospheric oxidation over practically the same temperature range. Their lubricity is like those of mineral oils, and many of the common effective wear preventive additives are also just as effective in diester oils. The diesters mix with mineral oils in all proportions. They attack the common types of rubber packings and

Table II—Viscosity-Temperature Characteristics of Polymer-Thickened Diester Fluids

Identification	Viscosity, centistokes								Kinematic V.I.	ASTM Slope
	210 F	130 F	100 F	77 F	32 F	0 F	−20 F	−40 F		
Di-(2-ethylhexyl) sebacate	3.32	7.96	12.6	19.5	60.9	187	445	1,410	154	0.71
Acryloid HF-845, 1.0%	3.96	9.3	15.0	23.2	70.7	217	540	1,640	185	0.67
Acryloid HF-845, 3.0%	5.59	13.3	21.4	33.1	99.3	307	740	2,320	182	0.61
Acryloid HF-845, 5.0%	7.58	18.3	29.4	45.5	137	436	1,060	3,180	173	0.57
Acryloid HF-845, 10.0%	14.5	37.2	58.6	92.1	290	894	2,100	.....	160	0.48
Acryloid HF-845, 20.0%	39.7	106	179	291	1,000	3,290	8,700	.....	143	0.41

Table III—Typical Properties of Standard Grades of LB-X Series Fluids

Standard grade	LB-140-X	LB-300-X	LB-550-X	LB-1200-X
Viscosity				
Centistokes at 210 F	5.7	11.0	18.8	39
Centistokes at 100 F	29.8	65.0	119	260
Centistokes at 0 F	1,270	4,000	8,800	24,000
Centistokes at −20 F	5,100	18,500	47,000	.....
Centistokes at −40 F	34,000	.....	.....	.....
Viscosity index	140	142	141	136
Pour point, F	−50	−40	−25	−15
Flash point, open cup, F	345	490	490	490
Fire point, open cup, F	410	585	600	610
Specific Gravity, 60/60 F	0.985	0.997	1.001	1.007

Table IV—Typical Properties of Standard Grades of 50-HB Series Fluids

Standard grade	50-HB-170	50-HB-260	50-HB-280-X	50-HB-400	50-HB-660
Viscosity					
Centistokes at 210 F	7.6	11.3	11.5	16.7	26.2
Centistokes at 100 F	36.5	56.2	60.6	86.6	143
Centistokes at 0 F	1,100	2,000	2480	3,500	6,700
Centistokes at −20 F	4,000	7,700	9,800	14,000	26,000
Centistokes at −40 F	20,000	45,000	.....	.....	.....
Viscosity index	156	153	150	149	144
Pour point, F	−45	−40	−35	−35	−30
Flash point, open cup F	410	410	500	430	430
Fire point, open cup, F	475	500	600	535	545
Specific Gravity, 60/60 F	1.030	1.039	1.047	1.046	1.053

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thermoplastic resins like polystyrene and lucite but not thermosetting resins like the phenolics.

By the addition of small percentages of polymeric V.I. improvers such as the polyesters, the diesters can be converted into more viscous oils with unusually high V.I.'s. An example is given in Table II of the effects of the polyester Acryloid HF 845 on the viscosity-temperature properties of the diester di-(2-ethylhexyl) sebacate (tradenamed Plexol 201). The use of 5 percent of the polymer results in an oil with a V.I. of 173, a viscosity of 29.4 centistokes at 100 F and only 436 centistokes at 0 F. Approximately 5 percent Acryloid HF-880 could also be used to thicken another diester, di-(3-methylbutyl) adipate to the viscosity requirements of the aircraft hydraulic oil specification AN-0-366. The addition of 10% of the same polymer would be needed to thicken di-(3-methylbutyl) adipate to meet the requirements of ordnance hydraulic specification 51-F-21. The resultant fluids would be much less volatile and flammable than the corresponding petroleum-base hydraulic oils, and the annoying tackiness problem would be greatly alleviated. However, their use would necessitate a change from the usual rubber packing materials.

Although diester-base instrument oils and ball bearing greases are now available from a number of commercial sources, none made up with polymers for hydraulics and other applications have yet been marketed.

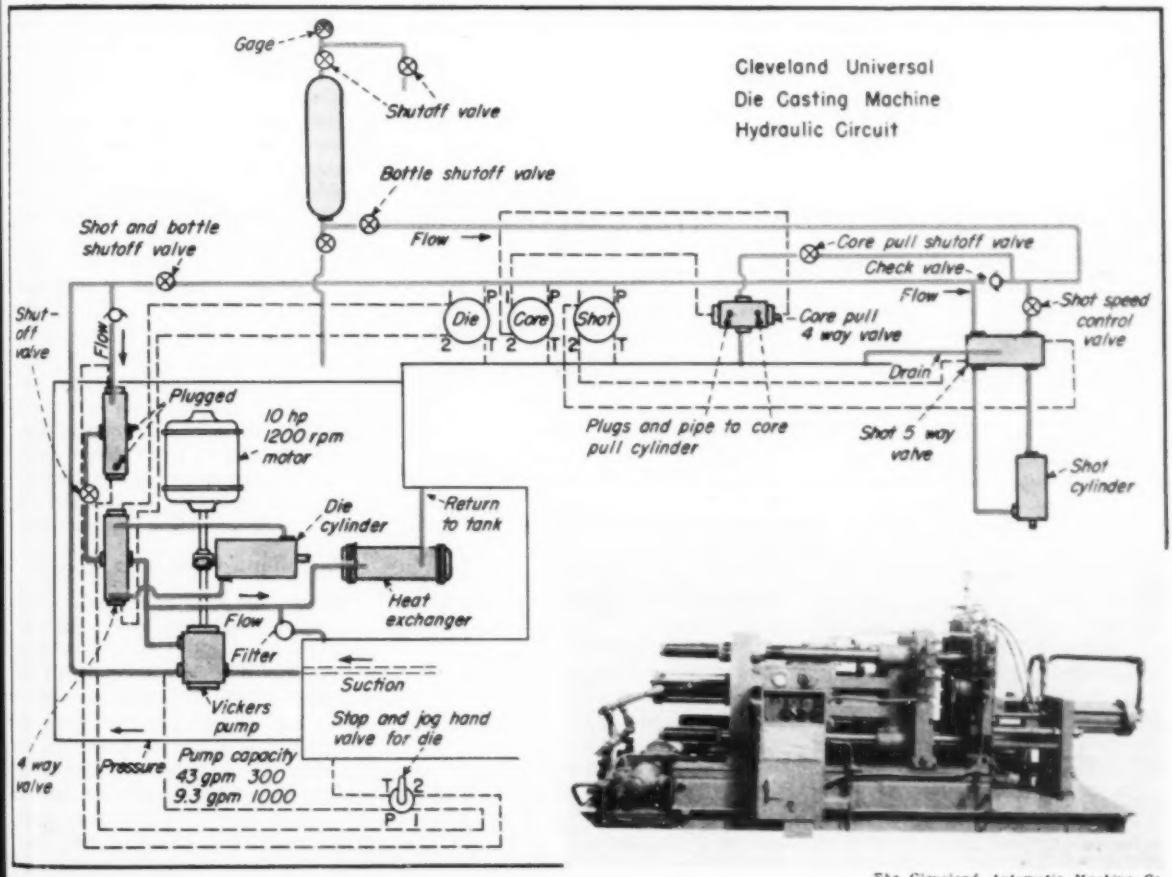
Polyglycols and Derivatives

A commercially available, non-hydrocarbon class of synthetics is the chemically related group of lubricants and hydraulic fluids marketed under the brand name "Ucon". These chemicals are polyethylene derivatives. Four series of these fluids are available; the LB and DLB series which are nearly insoluble in water, and the 50-HB and 75-H series which are very water-soluble. These fluids are available in a wide range of viscosities, with or without additives.

Although all four classes of "Ucon" fluids are useful as hydraulic fluids, the LB and 50-HB series are the most widely used, because of their very excellent V.I.'s, low pour points and availability. The DLB fluids, which have much better V.I.'s, are now available in pilot quantities. Tables III and IV show properties of typical fluids in the hydraulic range.

The commercial LB and HB fluids have flash points of approximately 325 F for a fluid of 10 centistokes (cs) at 100 F, up to 490 F for fluids with viscosities above 65 cs at 100 F. Their fire points range from 375 to 600 F. The S.I.T. ranges from 720 to 750 F, while the oxygen demand is 54 percent for the more viscous grades, decreasing rapidly to 20 percent at a viscosity of 10 cs at 100 F. These fluids are good lubricants and, as the oxidation products are either volatile or are dissolved by the fluid, difficulties with sludge formation are rarely encountered. The LB fluids are used rather than the 50-HB where water-insolubility is desired. They are also preferable where greater solubility in petroleum products is required. Table V shows the small swelling action of medium viscosity fluids on rubber.

The use of Ucon fluids in aircraft hydraulic systems has been limited because fluids having viscosities of 10 cs at 130 F have viscosities at —40 F of approximately 10,000 cs. However, their visometric characteristics



The Cleveland Automatic Machine Co.

Table V—Effect of "Ucon" Lubricants on Rubber

Rubber	Linear Percent Swelling		Condition of Immersion
	LB-285	50-HB-260	
Natural	2 to 3	−1 to 1	5 days, 160 F
Buna S	8 to 10	1 to 2	20 days, 212 F
Buna N	0 to −1	0 to 1	5 days, 160 F
Butyl	0 to −2	−2 to −3	20 days, 212 F

Table VI—Viscometric Properties of a Methyl Silicone and Military Specification Hydraulic Fluids

Viscosity Requirement	Specification			
	51-F-23 (Ord)	51-F-21 (Ord)	AN-0-366	Methyl Silicone
cs at 210 F min	8.0 min	10.0 min	5.1*	18.4
cs at 130 F min	25*	20*	10.0 min	34.3
cs at 100 F min	40 min	27 min	14*	44.9
cs at 0 F max	1500 min	215 max	94*	160
cs at −25 F max	7500 min	600 max	220*	250
cs at −40 F max	.....	.....	500 max	355

* Viscosity of a typical oil but not a specification requirement.

Table VII—Comparison of Some New Aircraft Hydraulic Fluids

Property	Petroleum Oil An-0-366	Hydrolube U-4	Phosphate Ester		Hexachloro-butadiene "A"
			"D"	"N"	
Viscosity, cs at 210 F	5.1	4.0	3.85	5.20	2.95
130 F	10 min	10	9.5	10.3	5.88
100 F	14	17	15.0	14.7	8.54
0 F	90	245	285	108	75
−20 F	165	610	950	195	160
−30 F	250	1000	2256	290	240
−40 F	500 max	1850	6000	442	425
Pour Point, F	−75 max	−60	< −60	< −70	< −65
Flash Point, F(C.O.C.)	200 min	None	360	305	(a)
Fire Point, F(C.O.C.)	215	None	470	365	(b)
S.I.T.	c.a. 500	850	> 1300	887	> 1300
Oxygen demand, %	12	> 65	23–25	54	> 85
Density, 68 F	0.80–0.85	1.06	1.08	0.98	1.5
Packing Materials (c)	Satis. (d)	Satis.	Poor	Poor	Poor
Wear Prevention	Satis.	Fair	Good	.....	Good
Shear Stability	Fair	Excellent	Good	.....	.....
Hydrolytic Stability	Satis.	Satis.	Doubtful	Doubtful	.....
Corrosiveness (72 hr at 212 F)					
Cold Rolled Steel	Satis.	Satis.	Satis.	Satis.	Satis.
Copper	Satis.	Satis.	Satis.	Satis.	Satis.
Aluminum 52S-O	Satis.	Satis.	Satis.	.....	Satis.
Duralumin 24S-T	Satis.	Satis.	Satis.	Satis.	Satis.
Cadmium plated steel	Satis.	Satis.	Satis.	.....	Satis.
Dow Metal C	Satis.	Poor	Poor	.....	Satis.
Dow Metal H	Satis.	Poor	Poor	Satis.	Satis.
Tackiness	Satis.	Satis.	Satis.	.....	Satis.
Surface Tension (dynes/cm)	28	31	32–33	.....	32
Foaming Test	Satis.	Satis.	.....	.....	Satis.
Rust Inhibition Test	Poor	No rust	No rust	No rust	No rust
Oxidation Stability	Satis.	Does not apply	.....	.....	.....
Miscibility with Pet. Oils	Complete	Immiscible	Complete	Complete	Complete

(a) Does not have a true flash point, burns around flame source.
(b) Fluid foams and boils at approximately 385 F.
(c) AN-P-79 rubber packings.
(d) Present packings designed for this fluid.

Table VIII—Properties of Three Commercial Hydrolube—Type Hydraulic Fluids

	Hydrolube "U"	Hydrolube "300"	Hydrolube "550"
VISCOSITY			
Centistokes at 130 F	10.0	34.5	70.9
100 F	16.5	59.5	119
20 F	116.5	485	990
0 F	247.4	1,213	2,378
−20 F	614	2,968	.....
−40 F	1,797	9,753	.....
Saybolt seconds @ 100 F	83.3	275	551
DENSITY, g/cc at 130 F	1.059	1.064	1.053
at 100 F	1.070	1.076	1.066
at 60 F	1.085	1.093	1.083
Freezing point, F	below −76	below −60	−15
Boiling point, F	211	209	209

can be improved without impairing their flammability properties by blending with high boiling aliphatic diesters.

Silicones

The liquid organo-silicon polymers or "silicones" are widely known for their remarkably high V.I.'s, low freezing points and oxidation stabilities. Because of these properties they have been particularly interesting for uses in hydraulics. (See article "Silicone Fluids in Hydraulics and Lubrication", PRODUCT ENGINEERING, Feb. ruary, 1949.) These synthetics are available in an extremely wide range of viscosities in several chemical varieties. The methyl silicones have the highest V.I.'s and lowest freezing points and are in general preferable for most hydraulic applications. The methylphenyl silicones are more suitable for applications where temperatures above 200 C (392 F) are encountered. However, this increased stability is obtained at the expense of a corresponding decrease in V.I. and an increase in the pour point. Caution is required in their use for such applications, as the silicones are inferior lubricants for steel sliding on steel.

"Stripped" silicones (volatile fraction taken out) are much less flammable than petroleum oils of comparable viscosities, and though they do not resist all five flammability tests, they are usually flame resistant.

Effects of silicone fluids on standard O-ring rubber packings are minor at ordinary temperatures, but as temperatures rise above 140 F, increasing troubles are encountered because of shrinkage and hardening. Associated with their abnormally low surface tensions (19 to 20 dynes per cm as compared with approximately 30 for petroleum hydraulic fluids) is the ability of the methyl silicones to creep readily over metal surfaces and through mechanical joints. Silicones are not compatible with petroleum oils.

In Table VI are given comparative viscometric data to show that a single methyl silicone fluid can fill the viscosity requirements of all high V.I. military hydraulic fluid specifications.

Phosphorus-Containing Fluids

Because of their high flash points and reputed flame resistance, the possible use in hydraulics of a variety of available phosphate esters have been studied. Aliphatic phosphate esters like trimethyl phosphate, which has a high phosphorous-to-carbon ratio, is significantly resistant to some fire hazards since the oxygen demand is 47 percent and the S.I.T. is approximately 900 F. Unfortunately, this compound has a flash point of only

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225 F, it forms acids in contact with water and it attacks the common types of rubber packings. The aromatic compound tricresyl phosphate which has been used as hydraulic fluid in die-casting machines in recent years is more fire-resistant and is stable, but has a low V.I. and attacks packings.

Commercial grades of tricresyl phosphate are high-boiling liquids (275 to 290 C at 10 mm Hg) having viscosities at 100 F of approximately 26 cs, flash points approaching 500 F, S.I.T.'s above 1,000 F, and oxygen demands varying from 19 percent to 85 percent, depending on purity. The pour point generally ranges from —10 to —15 F.

Two less flammable hydraulic fluids have been developed. One fluid, "N", is a mixture of an organic phosphate, a di-ester and additives. The other is a commercial aromatic phosphate ester-base fluid, identified as "D". The properties of these two fluids are given in Table VII. Both fluids will flash and burn when heated, "N" being the more flammable under these conditions.

Both the phosphate ester-base fluids "N" and "D" are miscible with petroleum hydraulic oils. Unfortunately, they also seriously attack the O-ring rubber packings used in existing equipment, and their adoption would require the replacement of nearly all existing rubber packings. Fluid "D" is reported to have excellent anti-wear characteristics and satisfactory shear stability. No information is available as to its tackiness, foaming tendency, or oxidation stability. Much industrial and government sponsored research is in progress on improved phosphorus compounds for aircraft hydraulic fluids, and further advances can be expected in the next five years.

Halogenated Fluids

All of the highly chlorinated fluids are nonflammable, are adequately resistant to atmospheric oxidation, but attack the commonly used rubber packings. Many chlorinated compounds evolve vapors when overheated or ignited and these probably are health hazards. However, recently there has been placed on the market a nonflammable hydraulic fluid "A" for aircraft, based on the use of hexachlorobutadiene and additives, having the properties listed in the last column of Table VII. This fluid is practically nonflammable to the fire hazards that may be encountered by commercial and military aircraft. It is miscible with petroleum-base fluids but there is still some uncertainty as to the toxicity in ordinary use or during overheating. The fluid is reported to attack some rubber packings but to be satisfactory with others. It is reported that the fluid performed satisfactorily in a 3,000-psi hydraulic system operating at 160 F. However, the volatility of hexachlorobutadiene would limit its usefulness at higher temperatures.

The wide use of existing chlorinated liquids in hydraulics will depend on the development and adoption of suitable packings and the observance of proper health precautions.

Fluorinated Compounds

Research programs on the synthesis and properties of organic compounds of fluorine were carried out during the war, and interest in the subject has continued to increase. The new compounds were studied for many possible applications, including dielectrics, plastics, stable heat transfer fluids, and nonflammable lubricants and hydraulic fluids. Hydraulic fluids made from completely fluorinated hydrocarbons (perfluorocarbons) are more stable than chlorinated hydrocarbons.

Although relatively few pure perfluorocarbons are available in quantity, enough properties have been studied to permit some generalizations. Mixtures of partially fluorinated hydrocarbons could be used with additives to develop nonflammable hydraulic fluids. Although their high pour points may limit the temperature range of application, fluorinated fluids may prove useful as base fluids for high temperature applications.

Water-Base Fluids

As water is an effective, cheap and stable fire-quenching material, it is easy to understand why it makes a "natural" base for non-flammable hydraulic fluids. Ethylene glycol appears to be the most suitable material for developing a water-base hydraulic fluid because it is one of the most efficient freezing point depressants, it is produced in large volume, and it is free from deleterious effects on common packing materials. It has a viscosity at 100 F of only 8.7 cs permitting the use of V.I. improvers; also an oxygen demand of 40 percent, and a S.I.T. of 856 F. Only when practically all of the water has been boiled off can such solutions support combusion. In fact, a 35 percent glycol-water solution is a good emergency fire-fighting fluid for use at subzero temperatures.

Lubricating properties of such solutions can be increased considerably by the addition of water-soluble polymers and wear preventives. As the variety of metals present in the hydraulic system is increased and as the requirements are lowered for the pour point, the problem of improving lubrication properties becomes difficult. Corrosion inhibition of aqueous hydraulic fluids for aircraft is more difficult than for engine anti-freeze solutions because of the greater variety of metals present. An additional problem is encountered with interference between many corroson inhibitors and wear preventives. In the construction of aircraft hydraulic systems, steel is replaced largely by aluminum (52S-O or 24S-T) and zinc- and cadmium-plated steel are often used to decrease electrolytic corrosion. Some aircraft components are made of high magnesium alloys, and this makes the corrosion inhibition problem even more complex.

Polymer-thickened, inhibited, glycol-water solutions prepared for hydraulic and lubricating uses were developed at the Naval Research Laboratory and given the generic name "Hydrolubes." Improved Hydrolube fluids and variations for special purposes are under development in several industrial laboratories. Hydrolube U-4, developed for aircraft use, performs satisfactorily in present hydraulic systems; however, the average pump life varies with the design. Gear pumps of manufacturer A can be operated for thousands of hours at 140 F and 3,000 psi, and manufacturer B's pumps, if modified by cadmium plating of the helical springs, also operate satisfactorily under these conditions of pressure and temperature. However, piston pumps of manufacturer C have shorter lives because of wear in one heavily thrust loaded ball bearing. Vane pump wear is also higher than desirable. This fluid has less effect on the synthetic rubber packings in use in aircraft than the present AN-O-366 petroleum hydraulic fluids. It causes even less deterioration of natural rubber packings, and this is advantageous in many applications. The resistance of Hydrolube U-4 to aircraft fire hazards has been fully confirmed by Civil Aeronautics Authority tests, and its utility has been verified in a number of naval planes which have been in operation for over three years with excellent results under all extremes of weather, speed and altitude.

Interest has grown rapidly for applications of these fluids on ships and submarines, and in a wide variety of industrial applications. One chemical manufacturer is already producing Hydrolube U for aircraft use and two more viscous grades for industrial use (see Table VIII). These fluids are being applied in die-casting machinery, manipulators for iron and steel mills, hydraulic presses, furnace controls, coal mining machinery, and hydraulic brakes. High temperature applications are limited by the boiling point of the solution.

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ON THE SPOT in this interview were Robert Hose, Henry Dreyfuss, and John Beinert. Among his clients Dreyfuss can list such concerns as Bell Telephone Laboratories, the RCA Victor Div. of Radio Corporation of America, Warner and Swasey Machine Tool Company, National Supply Company, and many others

The Role of the INDUSTRIAL

PE—Here's an embarrassing question as a starter, Mr. Dreyfuss. Why should a company with an engineering design department come to an industrial designer for aid?

DREYFUSS—It's not an embarrassing question. Many of them do not need to. Some of the larger companies have excellent industrial design departments. But the average company feels that the industrial designer practicing today brings a different perspective into the picture. He brings an entirely different viewpoint. He is in constant contact with many different things and is concerned with the development of a large number of widely different products. All that would be difficult to have in a company unless it were very large. Of course, companies such as General Electric and Sears-Roebuck make so many varied products they can afford industrial design departments which in themselves are separate entities and almost resemble an office like ours.

PE—Your business is, therefore, concerned primarily with small companies?

DREYFUSS—Not small companies, but companies with a limited number of products. A company can be large and still make only a limited number of products, in which case we can do a real job for them. When the number of products is limited, the design engineers usually do not have the time to devote to research and development and testing of competitive products. Before we would design a television cabinet, we would first go out and line up every other competitive make on the market and do a thorough customer research job on it. We would then apply to that product our knowledge gained from this study, as well as experience gained by the design of many other products.

Now, let's take the smaller manufacturer, the one who doesn't make many varied products. You see, the design department at Sears or General Electric is concerned with as many varied things as we might handle in this office. They might handle anything from an electric iron to a locomotive in the General Electric industrial design department, just as we do here. But with the smaller manufacturer, that is not often the case. Remember that a designer's perspective on a given product is always contingent upon the scope of his work with other products.

PE—In other words, he gets an idea from one field that he can apply to another?

"A DESIGNER'S PERSPECTIVE is always contingent upon the scope of his work with products other than the one on which he is working."

HENRY DREYFUSS has some definite ideas on the sound approach to product design and on the role of the industrial designer:

- **Convenience of use should be the first consideration when designing a product**
- **Manufacturing equipment available in a company's plant or in the plants of its suppliers must be investigated before designing begins**
- **Form should follow function—not the reverse**
- **Not every company needs the services of an industrial designer**

Here are the answers by one of this country's foremost designers and two key men of his staff, John Beinert and Bob Hose, to some rather pointed questions put to them by *Product Engineering* editors. The answers, obtained during a round-table discussion, clearly define the role, as well as the relative status of the industrial designer in industry.

DESIGNER

DREYFUSS—Well, it is a feeling—an approach. I could walk you around this drafting room and show you a boy working on a silver spoon, who last week was designing the parts for a ship. We feel it is best to let people work on various things around here for that reason. It contributes something to his knowledge and his judgment.

PE—What is the primary consideration in designing any new product or redesigning any old product?

DREYFUSS—The approach—the intelligent approach to any industrial design project is to take the five points that are the catechism of a good industrial designer. They usually are a surprise to manufacturers. They are amazed about it because so many people feel that we are just interested in what a product looks like. That is a consideration, but it is not the first consideration. The vital factors are first, convenience of use, including utility and safety; second, ease of maintenance; third, cost of manufacture and distribution; and fourth, merchandising and competition. And last but not least, we consider appearance. We apply what knowledge we have of form and line, proportioning and color to the other four factors. The fact that the designer studies and analyzes sales and distribution seems to surprise many people. We follow those things very closely. Why do people buy something? Sometimes if we know why they buy an alarm clock, it tells us why they buy a vacuum cleaner or a turret lathe or select a particular train or ship on which to travel.

PE—So appearance and cost, while not being first considerations, are still very primary considerations—is that right?

DREYFUSS—No question about it.

PE—Is where you start a design?

DREYFUSS—Well, technically it is not where we start, but it certainly all revolves around that. The first thing that we do before starting a design is to go to the client and find out what the limitations are of his factory: material-wise, tool-wise, what can he produce? We try to accept those limitations as a challenge and as a restriction. Then we go out and try to find out what competition is doing, because sooner or later, if we are doing a typewriter, that typewriter is going to be lined up on a table with a half dozen other typewriters. They are all going to be made by well-known companies. They are all going to cost within a few cents of one another. They are represented in a magazine with a full-page ad. So they are equal. They all perform well. They are all guaranteed. Why does a person buy one instead of the other? If we find out, I can take that same thing and apply it to alarm clocks, a vacuum cleaner, almost anything.

PE—Well really, then, when you start, you have to set up specifications for a product, is that right? It must

FIG. 1—**THE PROBLEM** is the redesign of a vacuum cleaner for The Hoover Company. Basic functional requirements are discussed and specifications established with client engineers.

FIG. 2—**PRELIMINARY SKETCHES** establish general appearance. Contour is lowered, weight decreased, color changed, and provision made for easier insertion of other attachments.

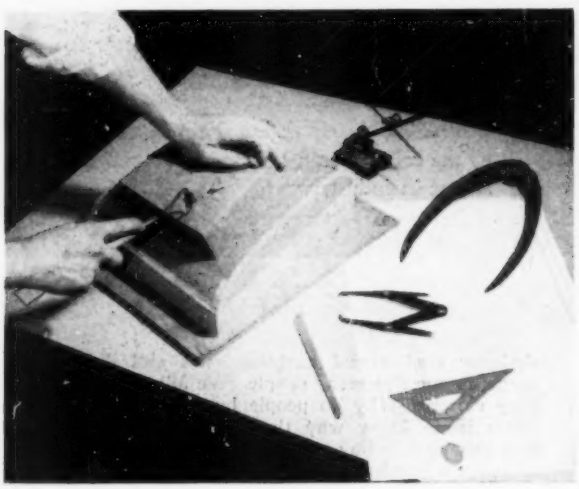

FIG. 4—**A CLAY MODEL** is the next step. Appearance and clearances, both of which are of importance, need not be visualized—they can be seen—in a three dimensional model.

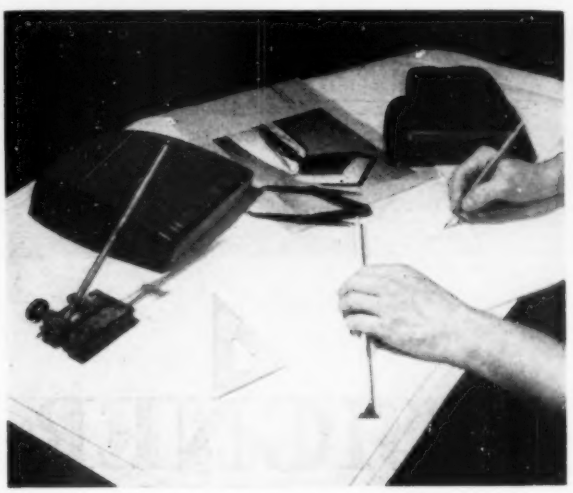

FIG. 5—**CHECKING CONTOURS** on the final model. By this time the major problems have been solved, only details need be checked prior to final tooling for quantity production.

be made this way, or that way, or this way, depending on the limitations of the equipment in the plant . . .

DREYFUSS—Oh, we might very well say to a manufacturer, "Look, you ought to be making this out of plastic," and maybe it would be worthwhile for us to experiment with some plastic. Maybe he doesn't go out and buy a lot of plastic equipment right off the bat, but he can buy production on the outside.

PE—If you took something like the Hoover vacuum cleaner, for example—if you started with that, would you set up a series of things you would like to accomplish by a redesign?

HOSE—Yes, we would. Now, we've worked for the Hoover Co. for many years, and are really a part of their organization. We know their future thinking pretty well. And if they are going to start a new, bigger program, we are brought in at the very inception, when it is just conversation. How big is it going to be? What market are they hitting for? What type are they going after? And I'll sit in to take it in with the other people—with the sales department, and the advertising, manufacturing, production, and engineering departments, and they then put the components together—what they think they want to put into that cleaner. From then on, working very closely with the engineering group—the Hoover engineering group—we start devising a cleaner. We'll take the component parts in three dimensions and cover them with a clay or plastic, you see, because I believe that everything should be designed in three dimensions as soon as possible. Get it off the drawing board.

PE— So you can physically see . . .

DREYFUSS—So we can physically see what's happening. We can see what clearances we have. Then immediately, I'd say hand-in-hand with all this technical information, we start thinking about the consumer—the person who is going to use it.

PE—Do you actually question the consumer or user?

DREYFUSS—We do on occasion. Or we ask our client to question the consumer, or we suggest, if it is a big

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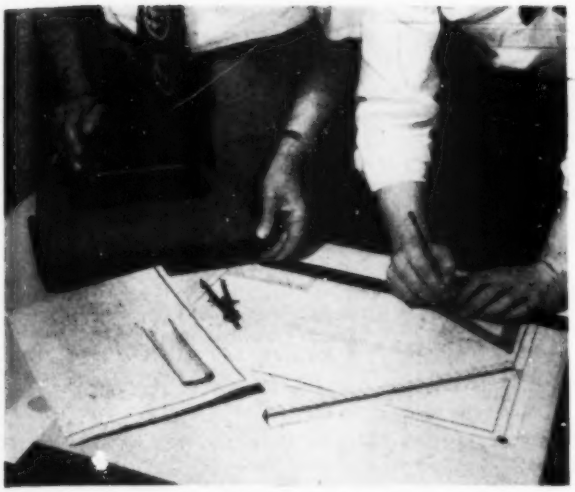

FIG. 3—TENTATIVE DRAWINGS are made to scale. Tolerances and contours are checked to determine if they can be made without manufacturing difficulty and at reasonable cost.

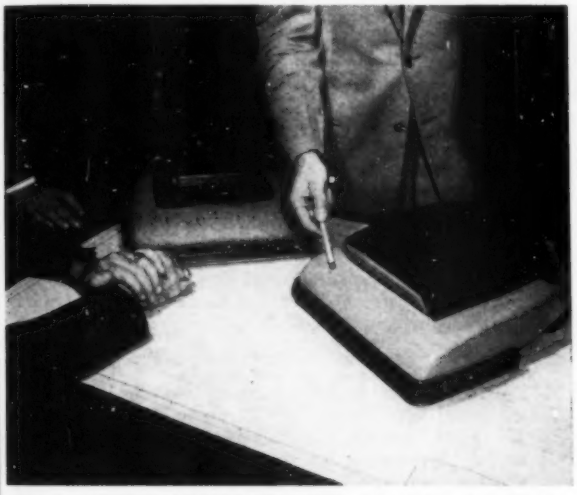

FIG. 6—THE OLD AND THE NEW. Note that the material and finish have been changed, wide angle lighting lens added for better visibility and the overall appearance is improved.

operation, that our client go out and get a research organization. And we will put that research organization to work making a market analysis.

We find out many things that way before we start. How can we price the cleaner? How long should the cord be? What is the most comfortable height handle to have? What color should it be? By changing a color just this year on a cleaner, we have advanced its consumer acceptance considerably.

PE—So you really are setting up specifications. For example, that cleaner can't weigh more than a competitive cleaner?

DREYFUSS—Oh, weight is very important. And decisions are based on many considerations—the technical requirements, the time factor, the cost factors, the manufacturing limitations, and the promotional characteristics. All of those factors are weighed constantly throughout this entire development, right up through the last stage where we are working on a small detail.

PE—Let's take appearance, Mr. Dreyfuss, and follow that right through. How do you achieve these various things that you want to achieve as far as the appearance of the thing is concerned. For example, in some products you want bulkiness. In others you want the appearance of movement. And how about color and texture which you mentioned—how do you go about deciding such things?

DREYFUSS—If I tell you that, you will be an industrial designer. There are no tricks, there is no magic to it. I suppose it has a lot to do with experience. We know, for instance, that certain textures are harder to clean than others, and we try to avoid them. On the other hand, we know that certain rough textures help manufacturing because they hide defects, and we try to adjust that.

But mostly, here the consumer comes into the picture. And our research—rather than going out and asking a lot of questions from door to door (I often feel by doing that you get strange answers because people answer what they think you want to know) we go right to the store or distributor who sells the product. Recently when I was designing an alarm clock, I went to a drugstore over on Third Avenue, and persuaded the proprietor to let me stand behind the counter and sell an alarm clock. I showed the first woman who came in and who wanted an alarm clock two models. One was the one that I had designed—a lightweight model—and one was a competitive make. She took one in each hand and weighed them, one against the other. I thought "Now—I really am on the right track, because she is going to take the one that weighs the least." Actually, she took the other because she thought she was getting more for her money in the heavier model.

Now, here is a clock that we are presently redisigning. There are two models—one of them is smaller and the other is larger, and they are in wood at present, but very shortly they will be in production in metal. The next thing we designed was the dial, and we went to an eye specialist to determine if it was as legible as could be made. That is an important item and one that attracts a potential customer to a product. Then we turned the clock around and started on the knobs. The problem here was to design a knob that could be used by a laborer with heavy, stubby hands, and a stenographer with thin fingers and large fingernails. You can see that they represent two extremes, and it was quite a problem to come up with a design that would be satisfactory to both. In other words, we tried to put ourselves in the customer's shoes to determine what they would want.

We recently redesigned the keyboard for the Royal Typewriter, using a key form on which it is impossible to break a fingernail. That was one of the primary complaints we heard from stenographers on making a check on what they liked and did not like about a typewriter.

PE—On a thing like a machine tool, would you consider the ultimate user or would your first consideration be the purchasing agent?

DREYFUSS—Our consideration would be, in the short run the purchasing agent, and in the long run the user. In other words, we would try to design a machine that would meet all the requirements of a purchasing agent and would impress him in so far as function, appearance, and usability were concerned, but our long range objective would be the men in the shop who would use the machine.

Recently we redesigned a machine at which the operator sat down. Normally a machine is designed so that an operator stands in front of it, but at this machine he actually sat in front of the machine while it was operating. The problem was to place the knobs

so that he did not have to reach too far for them and wasn't uncomfortable when he was reaching for them.

JOHN BEINERT—Actually we considered the design of the machine from a productivity standpoint. In other words, we tried to design the machine so that the maximum productivity could be obtained with minimum effort. Safety and maintenance were also factors that entered into the design. In short, what we tried to do was to fit the machine to its function.

PE—Let's assume that there are certain variations that could be made in the outward appearance of the machine after all functions have been considered. Are there any basic shapes that you try to put together to arrive at the best or most pleasing appearance?

DREYFUSS—No, we do not have any basic shapes that we try to follow. As one of the foremost architects once said, "form follows function." Consider a ball, for example. It represents the perfect shape for a rather complex device. It is thought of as a very simple element, now, but actually it represents the perfect example of the fact that the shape of the part should follow the purpose for which it is designed.

HOSE—Actually there is one thing hinted at in his conversation, and that is simplicity of form. We do not try to put in a lot of knickknacks and gimmicks on a product as used to be done by designers in the past.

DREYFUSS—In so far as color is concerned, office equipment represents a good example. A few years ago, all office equipment was black. Now you will find that typewriters, dictating machines, and other equipment are gray. There is a good reason for that, too. We found that if there is a shiny black surface, light is reflected from the surface into the eyes of the operator. This induces fatigue and decreases efficiency.

PE—Well, how do you go about selecting a color. Would you use red on a product, for example, to attract customer attention?

DREYFUSS—My answer, would be a definite "No". To attract a purchaser is one thing—but to consumate a sale is another. Experience proves that the consumer is usually conservative in his color selections and unless the product requires a brillant hue, we lean toward muted shades. The softer colors, gray, green, maroon and brown are better, and blue is coming into the picture. There are many examples where we changed just the color of a product and increased its sales.

PE—Sometimes you want a feeling of solidness. How do you go about it? Do you take off its legs and lower the profile of the product?

DREYFUSS—We would never take off the legs because of appearance. If the legs do not have a function, or dirt collects under them, we might take the legs off and drop the machine down to the floor.

BEINERT—There is some freedom, however, in establishing the outline of the product, and we try to work within the limits of the product's function to achieve a pleasing design.

PE—Do you use name plates or escutcheons to attract attention to the product in any way?

DREYFUSS—The name plate is something that has to be on a product to identify the manufacturer, but we do not usually use it as an attention-getting device. We realize that it has to be put on and we try to come up with an attractive looking identification after the main form of the product has been designed.

PE—Is the next step to select the material for the product?

DREYFUSS—Oh no. That is one of the first steps we take, the shapes that can be obtained will be governed to a large extent by the material. For example, we know that we can do certain things with plastics that cannot be achieved with metal, so it is necessary to select the material first.

PE—Do you do much research on materials?

DREYFUSS—Oh yes, considerable research on our own, and of course we have material suppliers who are very anxious to give us information. We rely also on the engineering departments of our clients.

PE—Do you provide for substitute materials when you design a part?

DREYFUSS—If by substitute materials you mean the same type of materials from various suppliers, we do. We specify a particular plastic on a part, but we may leave it to the client to obtain it from a supplier who can give him good delivery.

PE—Would you design a product that could be switched from die casting to plastics or stamped parts?

DREYFUSS—No, because there are certain limits on all of those processes that establish the shape of the product. We try to select the material and process first on the basis of the shape that the function of the part dictates it should have, and then on the basis of cost and manufacturing equipment available.

PE—I suppose that the anticipated production rate of the product enters into the picture during the selection of the materials.

DREYFUSS—Yes, we try to base the design on the equipment in the manufacturer's plant and the number of units that he expects to sell. Getting the product into production, however, is a problem that we usually leave to the manufacturer.

PE—Would you attempt to redesign a product or make minor corrections in it for better productivity or better use of materials if the production rate doubled.

DREYFUSS—If the anticipated production jumped and substantial costs could be cut by redesign, that might call for a major redesign job.

PE—Does cost analysis come into the picture at this point?

DREYFUSS—Cost analysis comes into the picture all along the line. Before we start any design, we try to determine from the manufacturer how much he expects to sell the product for, and we select the components and try to design to the cost limitations he imposes.

PE—On something like a machine tool, would you go into elaborate cost analysis methods?

DREYFUSS—It is my experience that cost is important in any product, regardless of its selling price. Of course, on a machine tool, the problem is not so acute. You do not have to try to shave pennies, as you do on an alarm clock, because the selling price of the machine is so much higher, but cost enters into any design.

PE—If a manufacturer comes to you with a product, is it normally quite expensive for him to retain your services?

DREYFUSS—I don't think we can give you figures that would be intelligent, unless we had a part to consider. I have had people talk to me and say "We don't think we can afford your services", and I answer that they cannot afford to be without a designer's services. Because, after all, a designer's fee is a drop in the bucket compared to the tool cost, or his advertising campaign. I like to think it's an insurance that he is going to get the properly designed product, and all of the facets that we talked about today are contained in the product. You can put it down as the same assurance a woman has if she is well dressed. He can then go out on the market with his product and enter into competition, knowing that it will sell.

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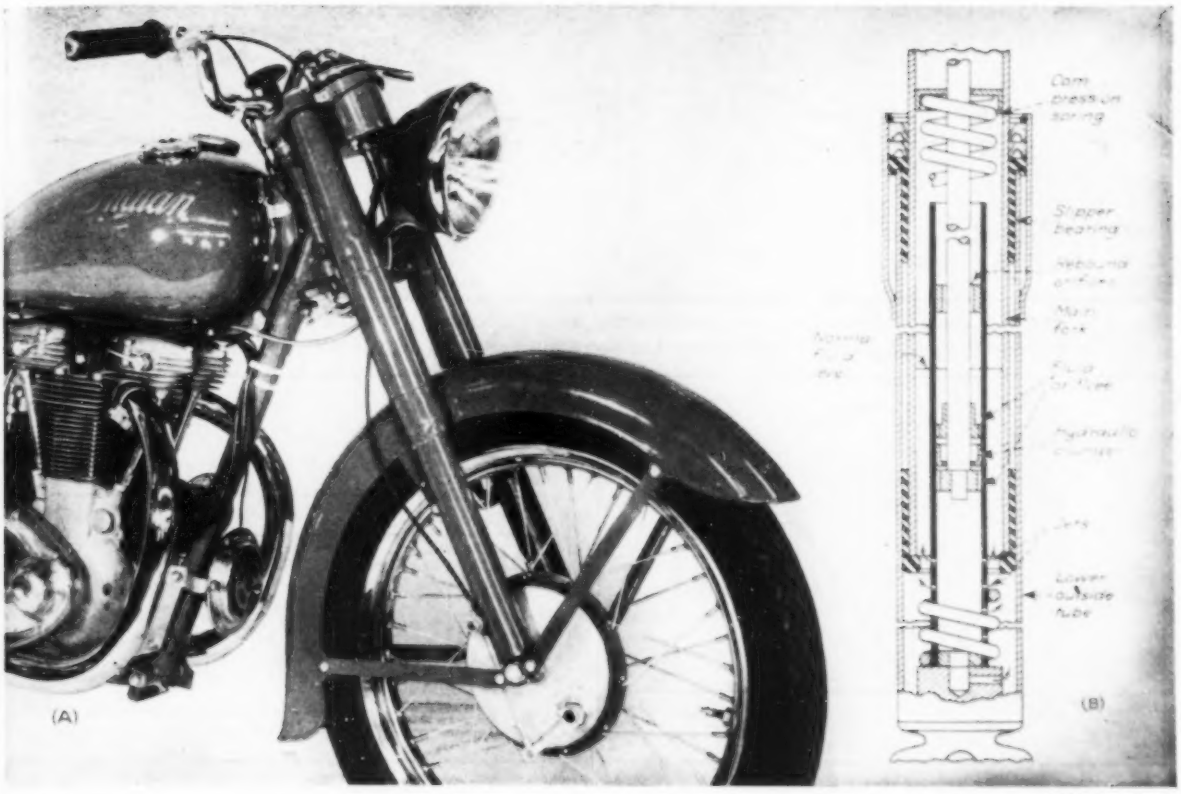

Indian Sales Corp.

HYDRAULIC METERING DEVICES are often used to reduce the effects of shock loads. (A) Redesigned motorcycle with a hydraulic-dampened telescoping absorber has proven more comfortable and easier to handle on rough roads. (B) Cross section of front fork shows details; motion of the hydraulic plunger is controlled by orifices and jets which cushion impact forces.

Dynamic Effects Of Load Application

The function of constant acceleration devices is to limit the maximum dynamic forces that are imposed on a structure, yet sometimes they cause higher peak loads. How can these momentary, peak loads and stresses be calculated?

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MANY DESIGNERS attempt to incorporate constant acceleration or constant force devices in their machines to reduce the effect of imposed dynamic loads. However, the usefulness of these devices can be overshadowed simply by a rapid application of force. There are many ways of altering the rate of loading. Two typical examples will be given to illustrate the effect of dynamic forces with the analytical solutions of these problems. In addition, the effect of a sudden application of force will be discussed. These methods should illustrate the importance of dynamic loads and be helpful to the designer in the solution of similar problems.

The problem of load application is a serious one, and every effort should be made to keep the rate of loading to a minimum. In this way, peak loads and peak stresses will be reduced. Under these conditions the resulting stress should be much lower. This design approach should reduce the material and weight of structural machine members.

In the recoil mechanism of large guns, it is common practice to design orifices in the hydraulic dash-pot to produce a constant retarding force for the full length of stroke. On this basis, a maximum amount of energy is supposed to be absorbed in a given stroke,

** The opinions and assertions contained herein are the private ones of the writer and are not to be construed as official or reflecting the views of the Navy Department or the Naval Service at large.*

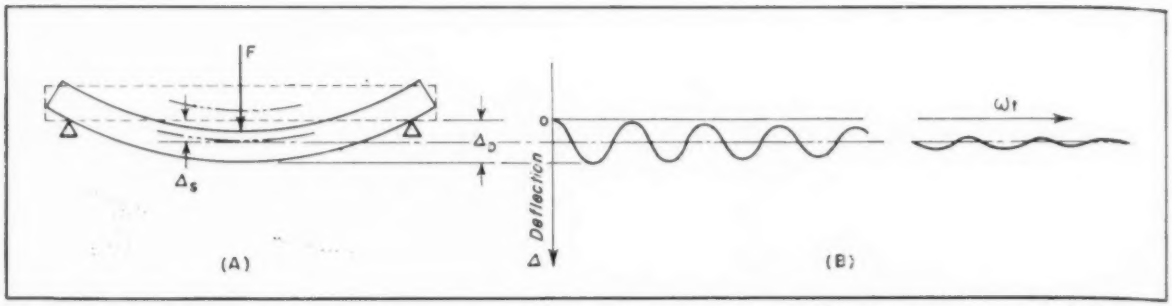

Fig. 2—Sudden application of force would momentarily deflect a simple beam (A) to position Δ_D. After a number of vibrations, (B) beam would settle to static position Δ_S. Beam would be alternatingly stressed much higher than if F were slowly applied.

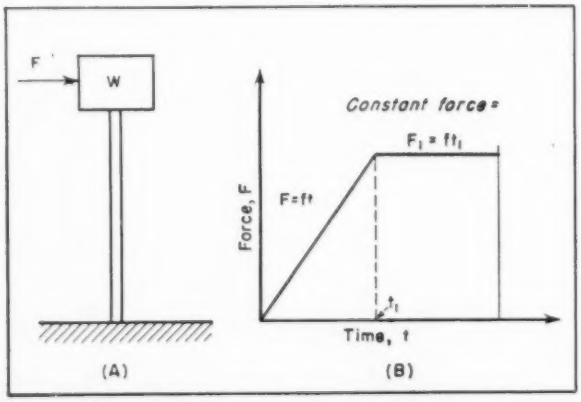

Fig. 3—Schematic diagram of gun mount. (A) Increasing force acts on beam. (B) Rate at which force is applied.

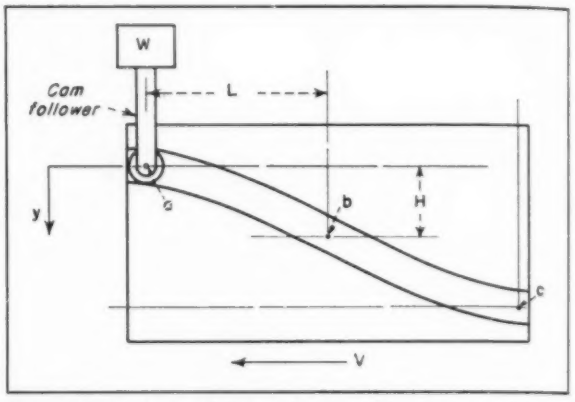

Fig. 4—Force on cam follower is affected by the type of curvature. Improper choice of curve can mean high dynamic loads.

thus producing the lowest forces on the structure. Actually, under these conditions the load is suddenly applied and momentarily the stresses become much higher than anticipated.

The case of curved railroad tracks also illustrates the problem of load application. Years ago, railroad curves consisted simply of circular arcs tangent to the straight sections. While the train was on the circular section, the centrifugal forces were constant; however, upon entering this curved section, both the train and tracks were

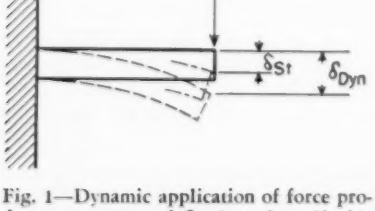

Fig. 1—Dynamic application of force produces a greater deflection than if this same force were applied gradually.

subjected to a suddenly applied load. To eliminate this severe loading condition, easement curves have been found to be necessary both for the entrance and departure from the circular path.

With an understanding of the problem of load application, two things can be done: (1) design the machine to reduce effects of the dynamic forces, and (2) calculate and compensate for the actual peak loads that occur. With these factors in mind, more effective mechanisms can be made. This design approach should result in an improvement of devices which are required to absorb kinetic energy such as aircraft landing-gear struts, automotive shock absorbers, and catapult buffers. In this way, the design of cam actuated devices can also be improved.

Dynamic Load Calculation

Since the internal stresses of a member can be expressed as a function of the deflections, it is most convenient to use the deflection to express the effect of dynamic loads. In addition, the severity of load application can be best stipulated by an amplification factor, A, which is the ratio of the deflection due to the dynamic force and the deflection produced by this force, if applied gradually. This relationship can be shown by:

$$A = \frac{\Delta_D}{\Delta_S}$$

where Δ_D is the dynamic deflection and Δ_S is the static deflection. If the load is slowly applied, then $\Delta_D = \Delta_S$ and this factor is equal to unity.

If a given force is suddenly applied to a cantilever beam shown in Fig. 1, the beam momentarily deflects to the δ_D position and then after a number of oscillations finally settles at the δ_S position. Under these conditions, deflections and also stresses occur which are much higher than those under static conditions. In this case, the amplification factor A is greater than one.

The value of the amplification factor depends upon the manner in which the external load is applied. For this reason, calculations of this factor will be made for three cases: (1) A suddenly applied load; (2) A load which increases proportionally with time; (3) A load applied at a harmonic rate.

Suddenly Applied Force

A simple beam, as shown in Fig. 2(A) is subjected to a suddenly applied force of F lbs. In this case, if W is the weight of the beam in lbs and K is the spring constant of the beam in lb per in. This beam would rapidly deflect to the position of $\Delta_S = F/K$. However, at this point the mass of the beam has acquired a certain amount of kinetic energy and so it continues to deflect until this energy is absorbed in the deflection to the extreme position of Δ_D. This beam would then have maximum potential energy and would spring back towards its original posi-

tion. This beam would vibrate until it comes to rest in the deflected position, Δ_s. This vibration is graphically shown in Fig. 2(B).

Expressed mathematically: the summation of forces acting at the center of the beam (which is deflected a distance of y) the differential equation of motion is:

$$m\ddot{y} = F - Ky \qquad (1)$$

where the mass, $m = W/g$ and the acceleration, $\ddot{y} = d^2y/dt^2$. The general solution of this differential equation is:

$$y = C_1 \cos \sqrt{\frac{K}{m}}\, t + C_2 \sin \sqrt{\frac{K}{m}}\, t + \frac{F}{K} \qquad (2)$$

For convenience, $K/m = a^2$. To determine the integration constants C_1 and C_2 the values of deflection, $y = 0$ when time, $t = 0$ are substituted in Eq (2).

Then,

$$C_1 = -\frac{F}{K} \text{ and } C_2 = 0;$$

The particular solution of Eq (1) is:

$$y = \frac{F}{K}(1 - \cos \omega t) \qquad (3)$$

From this equation the deflection is a maximum when $\cos \omega t$ is -1 (or when $\omega t = \pi$); under these conditions, the dynamic deflection $\Delta_D = 2F/K$. Since the static deflection $\Delta_s = F/K$, it is evident that the amplification factor, $A = 2$ for a suddenly applied load, and the effect on the beam would be to double the stresses.

Applied Force Increasing In Proportion To Time

In the case of recoil mechanisms of large caliber guns, the hydraulic shock absorber is designed with orifices to produce a constant load throughout the entire stroke. Fig. 3(A) will be used to illustrate the cantilever gun mount which has a large equivalent mass, $m = \frac{W}{g}$ that is subjected to the recoil force F. The recoil force is controlled to vary with time to point t_1, and then remains constant until the end of the stroke. This condition of loading is shown in Fig. 3(B).

In this instance there are two differential equations of motion; first, during the period that the applied force, F is increasing:

$$m\ddot{y} = ft - Ky \qquad \text{when } t < t_1 \qquad (4)$$

and second, during the interval that the force is constant and equal to $ft_1 = F_1$:

$$m\ddot{y} = ft_1 - Ky \qquad \text{when } t > t_1 \qquad (5)$$

The particular solution of Eq (5) has a larger value for the deflection y, than Eq (4). It is:

$$y = \frac{F_1}{K}\left[1 - \frac{\sin \omega t}{\omega t_1} + \frac{\sin \omega (t - t_1)}{\omega t_1}\right] \qquad (6)$$

when $t > t_1$

This equation has been simplified by the substitution, $\omega^2 = K/m$.

To determine the time, t_1 when the deflection is maximum, Eq (5) is differentiated with respect to t and the value of velocity $dy/dt = 0$ is substituted. As a result,

$$t = \frac{\omega t_1 + 2\pi}{2\omega}$$

this value of t substituted in Eq 6 gives the maximum deflection,

$$y_{max} = \frac{F_1}{K}\left[1 + \sqrt{\frac{2 - 2\cos \omega t_1}{\omega t_1}}\right] \qquad (7)$$

The severity of load application in this case can be determined by the ratio, $A = y_{max}/y_s$; where the deflection from a gradually applied or static load is, $y_s = F_1/K$. The amplification factor is:

$$A = 1 + \sqrt{\frac{\alpha 2 - 2\cos \omega t_1}{\omega t_1}}$$

From this equation the amplification factor is at its optimum value of 1 when $t_1 = 2\pi$, or 4π.

Force Applied At A Harmonic Rate

The harmonic method of load application is useful in many phases of automatic machinery. Fig. 4 illustrates a cam which is moving at a constant velocity V. In this case a comparison will be made between constant acceleration and harmonic motion. With constant acceleration $[\alpha]$ from a to b, the displacement, $[H]$ can be expressed by:

$$H = \tfrac{1}{2}\alpha T^2$$

since the time interval, $T = L/V$; this equation can be written:

$$\alpha = 2H\frac{V^2}{L^2}$$

Weight of the follower is W, and so the resulting dynamic force would be:

$$F_D = 2\left(\frac{W}{g}\right)\left(\frac{2HV^2}{L^2}\right) \qquad (8)$$

Since the constant acceleration force is considered as suddenly applied, a factor of 2 must be used. This factor was previously derived.

If instead, this cam were designed to produce simple harmonic motion, the vertical displacement would be:

$$y = H(1 - \cos \theta t) \qquad (9)$$

where: $\theta = \pi V/2L$

Differentiating this equation twice, the acceleration would be:

$$\frac{d^2y}{dt^2} = (H\theta^2 \cos \theta t)$$

Substituting this in the form of Newton's law: $f = ma$;

$$F_H = \frac{W}{g} H\theta^2 \cos \theta t \qquad (10)$$

since $\theta = \pi V/2L$ and using the maximum value of $\cos \theta t = 1$,

$$F_H = \left(\frac{W}{g}\right) H\left(\frac{\pi^2 V^2}{4L}\right) \qquad (11)$$

Comparing the effect of the forces: when suddenly applied, Eq (8) and applied harmonically, Eq (11) the amplification factor is:

$$A = \frac{F_D}{F_H} = \frac{4\,\frac{W}{g} H\,\frac{V^2}{L^2}}{\frac{W}{g} H\,\frac{\pi^2 V^2}{4L^2}} = \frac{16}{\pi^2} \text{ or } 1.62$$

This equation illustrates that the suddenly applied (constant acceleration) force produces a loading which is 62 percent higher than if the force were harmonically applied.

In this manner other cases may be analyzed to determine the magnitude of forces that may be imposed on a structure. Besides this, it should be evident that designing constant acceleration mechanisms for shock absorbing machines may not produce the lower stress which is desired but much higher peak loads and stresses.

* * * *

MAN'S FIRST VENTURE in the field of guided missiles occurred long before the development of the V-1 buzz bomb and the V-2 rocket by Germany in the last war. Actually, neither of these weapons were "guided" missiles in the modern sense of the word, because there was no means of determining and correcting errors in their course after they had been launched. Our own armed forces sponsored experiments with flying bombs and aerial torpedoes between 1916 and 1922. These missiles had many of the same characteristics of the later German variety.

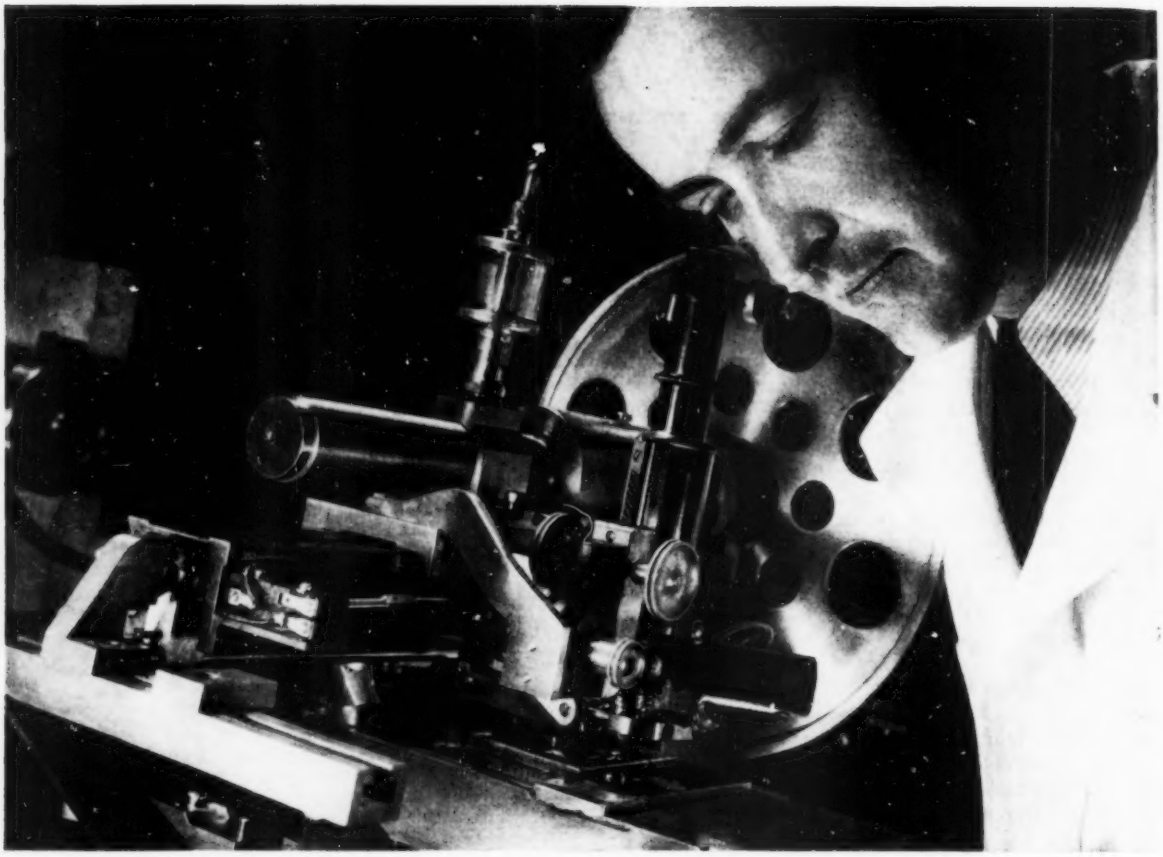

Fig. 1—To establish reproducible surface finish standards for manufacture of sliding parts, General Motors Research Laboratories has developed this machine which can rule up to 10,000 lines per inch in gold master blocks. The author is looking through microscope attachment at a specimen lying beneath the fine diamond cutter. Electroplated replicas of these surfaces are measured.

Surface Finish and the

Practical aspects of a new approach to the measurement, specification and application of surface finish control of metallic parts. Types of instruments, standards for various production methods, and physical properties relating to condition of the metallic surface are compared and evaluated.

SEVERAL BASIC CONSIDERATIONS should be recognized by the designer before placing controls on the surface finish of a part. The most important consideration is: "Does the surface in question need to have a controlled finish?" One simple answer is: "If the surface has ever had to be refined to obtain the proper operating characteristics, then the finish must be controlled.

If the design is new, the designer must fall back on previous experience or experimental data. He must ask himself if the surface is to be subjected to wear and friction—what type and how much. Is appearance a factor? Will the piece be subject to fatigue? Some of the effects of these factors on surface finish were discussed in the article "Surface Finish Control" by Dr. C. R. Lewis, which appeared in the August issue of PRODUCT ENGINEERING. See also "Surface Finish Definitions and Standards," Reference Book Sheets, p 163 August issue.

A second consideration is the cost of producing a desired finish. Often, a designer tends to "over-engineer" a surface, thereby increasing the cost with no significant increase in quality. The cost factor becomes less important when quantity production is anticipated.

How close to specify the surface is a third consideration important to the designer. Should he go so far as to tell the shop just what machine method should be used? In designating surface finishes, the designer should allow tolerances that are sufficient for practical machining operations. As for the degree of control, the designer should usually refrain from dictating the machining method to the shop. The shop supervisor has a better knowledge of the capabilities

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Fig. 2—Typical applications for various surface finishes. Because of the great variety of special conditions of use, these should not be considered mandatory for the applications indicated, but rather as the best average practice.

Designer

ROY P. TROWBRIDGE
General Motors Corporation

of his machines, and may be able to set up the work with less cost and fewer man hours by one method than another. However, where laboratory results or previous experience indicate that there is only one method of machining that will do the job, the designer should indicate this on the drawing by a suitable note.

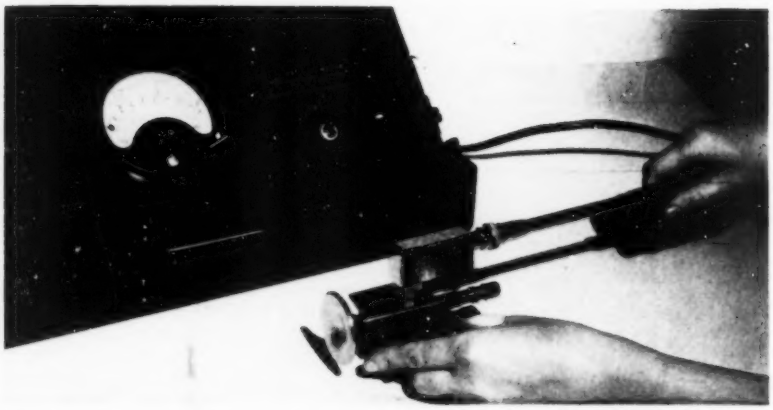
Where Should Surface Finish be Controlled?

Application of a specific surface finish on a specific piece does not imply that all pieces of a similar function should be subjected to exactly the same treatment. A comparison of the finish requirements on the piston pins in two currently popular automobile engines reveals a striking difference. One requires that the pins be lapped to four microinches rms roughness; the other is satisfied with a grind of only eight microinches. Despite the 100 percent disagreement in specifications, both piston pins give satisfactory service. This is possible because of the differences in general construction, which include type of bushing material, unit pressures experienced, linearity of loading, amount of lubrication, operating temperatures, and many other factors

It is difficult for any one authority to lay down any hard and fast rules for the application of surface finish because of the almost infinite variability of conditions that effect the application. James A. Broadston (see References) observes that designers can help this situation by making intelligent studies of surface finish applied to the products they design and by seeing that this data is made available to others through publication or technical papers. Specific applications are cited in this article, but they should not be construed as making the particular finish mentioned mandatory for all similar cases. Fig. 2 lists some typical applications of surface finish.

Finer surface finishes permit greater allowable loads for perfect conditions of hydrodynamic lubrication, since there is less likelihood of metal-to-metal contact through the thin film caused by high unit loading. However, the conditions for perfect lubrication seldom exist. Where only borderline lubrication exists specifying surface finish is more complicated. This is true in the cylinder of an internal combustion piston type engine. The rings, pressed tightly against the cylinder wall, receive only intermittent and scant lubrication. Since the optimum cylinder should be perfectly round and there should be perfect concentricity between the piston and cylinder, great care is exercised to perfect circular rings with even, radial pressures on the cylinder walls. One designer believed that additional wear service life could be gained by making the cylinder wall surface considerably smoother. The surface roughness of the cylinder walls was reduced from 15 to about five microinches average, and a different process was used to obtain the finish. Practical test work led engineers to believe that the improved surface finish was a great step forward. In the field, however, oil consumption was high from the start and never improved. The cylinders were easily scored, pistons were scuffed, and the rings did not seat properly. Investigation revealed that the geometric precision in production was not equal to that obtained in making the original test engines. Thus, with present production precision, the cylinder walls should have sufficient roughness for seating the rings and equalizing any small geometric imperfection in the processing. However, too coarse cylinder wall surfaces lead to excessive initial wear because of the rapid increase in clearance and the abrasive action of the worn away metal.

Improvement of a product is not always synonymous with refinement of surface finish. An example is the reduction gear scuffing experienced by Packard when making Rolls engines during the war. The frequency of rejects on these particular gears after the green run was sometimes as high as 30 percent. The original surface finish specification perpendicular to the pitch line of the gear tooth was 10 microinches rms. The gears were 57 to 59 Rockwell C hardness with Hertzian loading up to 200,000 psi. One of the causes of scuffing was that the surface finish on the gear teeth



Physicists Research Company

Fig. 3—Profilometer measures average surface roughness in rms microinches and is well adapted for shop use. Motion of stylus in tracer is relative to mean surface.

was too smooth and included too much waviness. To improve the surface finish, new roughness specifications were set at 15 to 37 microinches rms. After effecting these and other important changes on the gears and the method of run-in the scuffing was virtually eliminated.

When operating steel gears with bronze, surface finish considerations are quite different. For long service life of the steel-bronze combination, the steel gear should be as smooth as feasible, while the bronze gear surface can be relatively rough. If the steel gear is not kept smooth, abrasive action on the bronze gear leads to rapid tooth failure. This is true wherever hard and soft metals must operate together, such as in bushings and valve guides. The roughness on the soft metal should not be excessive after break-in.

Surface finish should be controlled where dry friction exists, as in brake drums, clutches, machine tool bits, threading dies, stamping dies and rolls. Uniformity and smoothness of operation of friction devices, depends on the surface finish of the metal. Service life of cutting tools and dies is increased by improving surface finish. In addition to service life, the appearance of work produced by a die is directly affected by the surface finish of the die. This is true also for extrusion and precision casting dies.

Surface finish control is also applied to parts subjected to fatigue stresses. For example, in one set of aircraft turbine gears, the tooth forms are hobbed and then shot blasted for fatigue resistance. The final functional tooth geometry is ground. But if the final grind so much as touches the root fillet of the tooth, the gear is rejected. Although not quite so critical, surface finish control is necessary on many highly stressed members of aircraft engines, such as connecting rods, studs, and fillets at changes of cross section. In aircraft fuselages, all tension members are usually checked for machine marks or flaws. In the automotive field, the fatigue problem is handled more by design than by surface finish control because the weight of parts is not as critical as in aircraft.

When close tolerance fits are required, as in injectors and high pressure hydraulic cylinders, it is desirable to control the surface roughness and waviness. Low limits on a tolerance may dictate the degree of surface finish, but usually the surface finish should be specifically designated. The surface finish of interference fits should be controlled if the loading on the joint is to be severe, because the degree of mechanical attachment depends directly on the surface roughness as well as concentricity of the parts. On contact fits, the degree of surface finish affects the heat transfer characteristics.

It should be emphasized that all surfaces do not require surface finish specification. Even though a piece is held to very fine tolerance, the degree of surface finish may be only an adjunct of the machining method. To designate all surfaces would detract

Brush Development Company

Fig. 4—Brush Surface Analyzer reproduces magnified trace of roughness profile on direct inking oscillograph. Average roughness meter is also provided. This instrument is adapted to inspection of microscopic surface irregularities on most metallic surfaces normally encountered in machining operations.

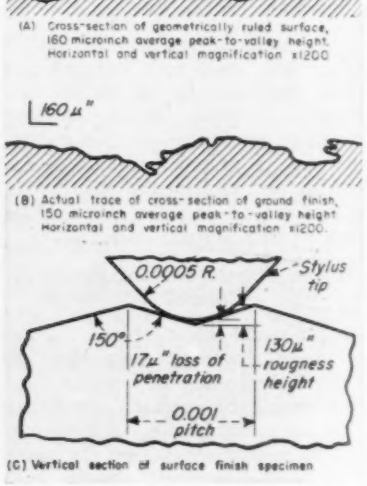

160μ"
(A) Cross-section of geometrically ruled surface, 160 microinch average peak-to-valley height. Horizontal and vertical magnification x1200
160μ"
(B) Actual trace of cross-section of ground finish, 150 microinch average peak-to-valley height. Horizontal and vertical magnification x1200.
0.0005 R.
Stylus tip
150°
17μ" loss of penetration
130μ" roughness height
0.001 pitch
(C) Vertical section of surface finish specimen

Fig. 5—(A) and (B) Geometrically ruled specimens show recurrent pattern. (C) Demonstration of deep penetration of stylus tip in geometrically ruled specimen.

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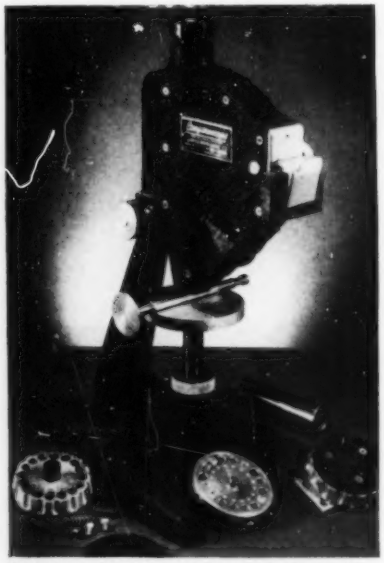
from those surfaces that do need control, and would unnecessarily raise production costs.

Measurement of Surface Finish

Before the advent of the profile measuring instruments, machined surface specimen comparators were used for surface finish control. However, because of the limitations of the human eye and sense of touch, the typical machined specimen sets were found lacking in accuracy.

STYLUS TYPE INSTRUMENTS. The present basis of surface roughness control is the stylus type surface measuring instrument. One of the instruments of this type is the Profilometer of the Physicists Research Company, Fig. 3, which has a diamond stylus mounted in a suitable tracer head. The motion of the diamond is measured relative to a mean surface established by two or more shoes riding on the crests of the adjacent surface roughness. By means of a small voltage created by the motion of a coil in the field of a permanent magnet, average roughness is read directly from a voltmeter. The Brush Analyzer (Brush Development Company), Fig. 4, is a similar type of instrument that measures motion of the diamond stylus relative to the mean surface by the absolute voltage created by the deflection of a piezo-electric crystal. Within their response bands, both instruments are very effective when properly calibrated. However, in use in the field, it is difficult to obtain uniform results between the two instruments and even between any two instruments of the same make.



Link Engineering Co. and Compar Instrument Co.

Fig. 6—Comparoscope compares pilot specimen with unknown by placing magnified images in juxtaposition. Specimens are viewed through a single eyepiece.

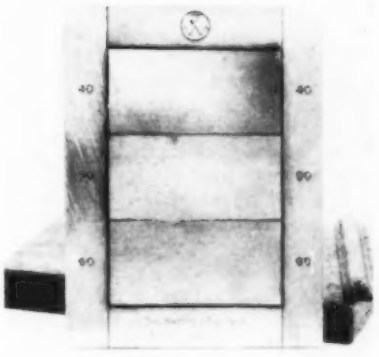


Fig. 7—Electroplated replicas of geometrically ruled standard specimens.

Bausch and Lomb Company

Fig. 8—Microinterferometer checks surface waviness by diffraction gratings.

The stylus type instruments have a definite cut-off point with regard to pitch or width of the roughness. The Profilometer response is linear up to roughness widths of $\frac{3}{32}$ in., the Brush Analyzer up to 0.025 in. If the roughness to be checked exceeds these widths, true readings will not be obtained.

The Profilometer was primarily designed for shop use and readings are taken by traversing the workpiece with a hand held stylus head. However, a moto-trace is available. The Brush Analyzer uses a motor driven tracer exclusively. In addition to the average roughness reading, the Brush Analyzer equipment includes a direct inking oscillograph which reproduces a magnified profile of the roughness. The oscillograph charts are helpful in studying the character of surfaces.

A recent development is a stylus type instrument known as the Proficorder, which has the added advantage of being able to measure waviness or long curves in a surface. The reference surface of the Proficorder is established by accurately trued ways on which the pick-up head travels. The Proficorder produces a chart record of the surface from which both roughness and waviness can be determined.

TAPER SECTIONING. A more absolute method of determining surface roughness is by taper sectioning, described in the article by Dr. Lewis in the August issue of PRODUCT ENGINEERING. However, since it destroys the surface, taper sectioning is of value to the researcher, but is impractical for production use.

GEOMETRIC SPECIMENS. Because of the lack of agreement between the various instruments, General Motors Research Laboratories and Chrysler Research Section have combined their efforts with electroforming experts to produce a series of flat $\frac{3}{4}$ by $1\frac{1}{2}$ in. specimens, having roughness values in agreement with preferred values given in the SAE Surface Finish Standard (page 163, August PRODUCT ENGINEERING) from 250 microinches down to 5 microinches average roughness height. Produced by ruling lines in a flat gold surface with a diamond tool of known cross sectional angle, forming a recurrent vee shaped pattern having an equal angle and height for each groove, Fig. 5 (A). The relative uniformity of the geometric specimens can be seen in Fig. 5 (B). Fig. 5 (C) demonstrates how the wide angle of the geometric pattern allows deep penetration of the stylus tip.

Electroplated replicas of these surfaces are measured by taper-sectioning. The complete set of specimens is mounted in a master fixture and all subsequent replicas are reproductions of this fixture. One of these reproductions is shown in Fig. 7. Experimental data proves the geometric specimens to be ten times more accurate than the typical machined surfaces. It is felt that wide adoption of the geometric specimens will correlate all of the instruments and typical machined specimens, by providing an invariable yardstick.

OPTICAL METHODS. As an adjunct to visual comparison of surfaces, comparison microscopes and reflection type comparators have been designed. Both of these types must be used with calibrated pilot specimens of the work being checked. The Comparascope, a comparison microscope (Fig. 6), compares the workpiece with a calibrated roughness of similar processing on a split field, both surfaces being viewed through a single eyepiece. Reflection

type surface comparators check the reflectivity of the unknown surface against an approved pilot specimen of the workpiece. The Merz Pico and Comtor Comparator are instruments of the latter type. In both types the pilot specimen must be approved by an instrument reading in micro-inches, if the surface to be controlled by standard preferred roughness values. These instruments are considerably cheaper than the stylus type, and although their readings are not absolute, they are satisfactory for production runs.

Until the introduction of the Profi-corder, the optical flat was the only accurate means of checking small waviness on flat surfaces. If the optical flat is placed in contact with a wavy surface of sufficient reflectivity in the presence of nonochromatic light, light and dark interference bands from a contour map of the surface, from which total height and frequency of the waviness can be determined. A refinement of this principle is used in the Interferometer and the Micro-interferometer (Bausch and Lomb) shown in Fig. 8. The main disadvantage of this method is that the surface must have sufficient reflectivity to form the interference bands. Roughness can be determined with the Micro-interferometer up to depths of approximately five times the wavelength of the monochromatic light. Beyond this, the pattern becomes too confused.

Dark and light field microscopy are also good methods of qualitative laboratory investigation of surfaces. For quantitative analysis, Schmaltz (see References) developed the optical cut method of viewing a surface, while Way (see References) adapted the principle in his straight edge method. In the former, a cross section of a surface profile is obtained by projecting the image of a light slit on the surface at a controlled angle of incidence, and viewing the reflection through a vernier microscope. The latter method is to project the light at a controlled angle onto the surface over the edge of a razor blade; the resultant shadow is studied microscopically. These methods are more adaptable to coarse surfaces—the lower limit is about 30 microinches peak to valley.

OTHER METHODS. For production comparison of surface roughness, an air gage method is developed by the Sheffield Gage Company. In this method a constant pressure source of air escapes past a thin metal disk in contact with the surface. The metal disk is backed by rubber, which conforms to the surface. The discharge of the air is mainly a function of the roughness of the surface. Magnifications up to 60,000 times the peak-to-valley roughness have been obtained. The gage must be suitably calibrated before readings can be quantitative.

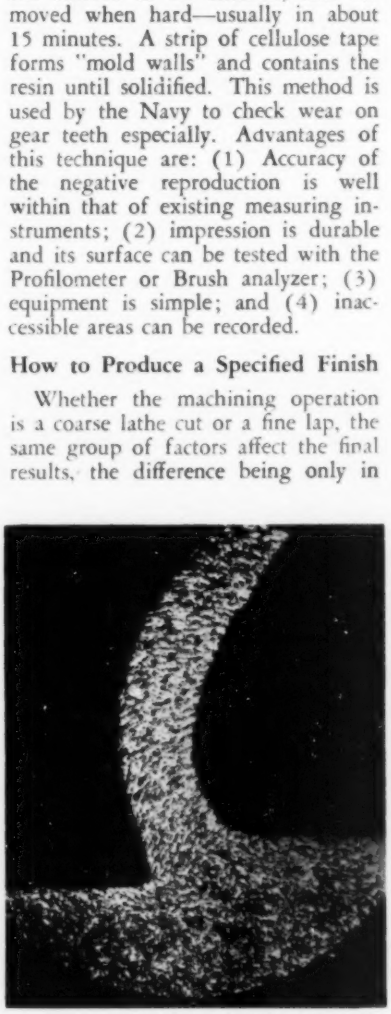
Another qualitative method of studying surfaces is the use of a plastic replica of the surface. The Fax Film Company markets a set consisting of solvent, transparant acetate tape, holders and a comparison viewer which magnifies the replicas 30 times. Impressions of unknown surfaces can be stripped off and compared with approved surfaces, and accurate comparisons may be made of surfaces before and after wear. This method is relatively inexpensive.

A similar method is the use of a liquid casting plastic, of the polyester type, which is supplied as a two-or three-component kit containing resin, filler and catalyst. One kit of this type is marketed by Marco Chemicals, Inc. The catalyzed resin is poured against the surface to be checked, and removed when hard—usually in about 15 minutes. A strip of cellulose tape forms "mold walls" and contains the resin until solidified. This method is used by the Navy to check wear on gear teeth especially. Advantages of this technique are: (1) Accuracy of the negative reproduction is well within that of existing measuring instruments; (2) impression is durable and its surface can be tested with the Profilometer or Brush analyzer; (3) equipment is simple; and (4) inaccessible areas can be recorded.

How to Produce a Specified Finish

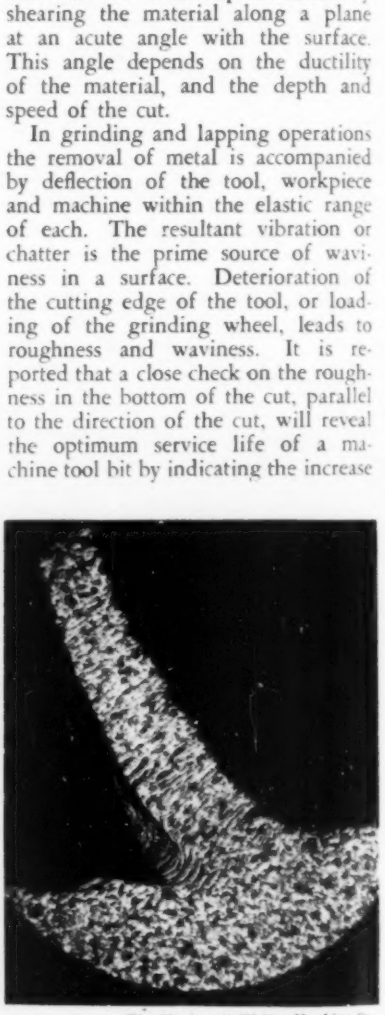
Whether the machining operation is a coarse lathe cut or a fine lap, the same group of factors affect the final results, the difference being only in the degree of coarseness. With ductile materials and tool bit cutting, the ideal chip formation for the best surface finish is the "type 2" or continuous chip without the so-called built up edge (Fig. 9). For non-ductile materials, the best surface finish is obtained when the pitch of the discontinuous or "type 1" chips is small. Roughness of the surface in these two types is affected by the depth of the cut, the sharpness of the tool, and the progression. Quite often, however, the ideal conditions of chip flow do not exist, then the "type 3" chip, having a built up edge, is formed (Fig. 10). Fragments of the built up edge remain as part of the surface, accounting for added roughness in the groove of the cut. Factors affecting formation of the built up edge are, in addition to the aforementioned, speed of the cut, angle of rake, temperature of the workpiece, polish of the tool bit, lubricant, and tool material. The chip is formed by shearing the material along a plane at an acute angle with the surface. This angle depends on the ductility of the material, and the depth and speed of the cut.

In grinding and lapping operations the removal of metal is accompanied by deflection of the tool, workpiece and machine within the elastic range of each. The resultant vibration or chatter is the prime source of waviness in a surface. Deterioration of the cutting edge of the tool, or loading of the grinding wheel, leads to roughness and waviness. It is reported that a close check on the roughness in the bottom of the cut, parallel to the direction of the cut, will reveal the optimum service life of a machine tool bit by indicating the increase



The Cincinnati Milling Machine Co.

Fig. 9—Type 2, or continuous chip without built-up edge. The flow of metal is not greatly retarded by friction.



The Cincinnati Milling Machine Co.

Fig. 10—Type 3, or continuous chip with built-up edge. Frictional resistance to flow of chip along face of tool is great.

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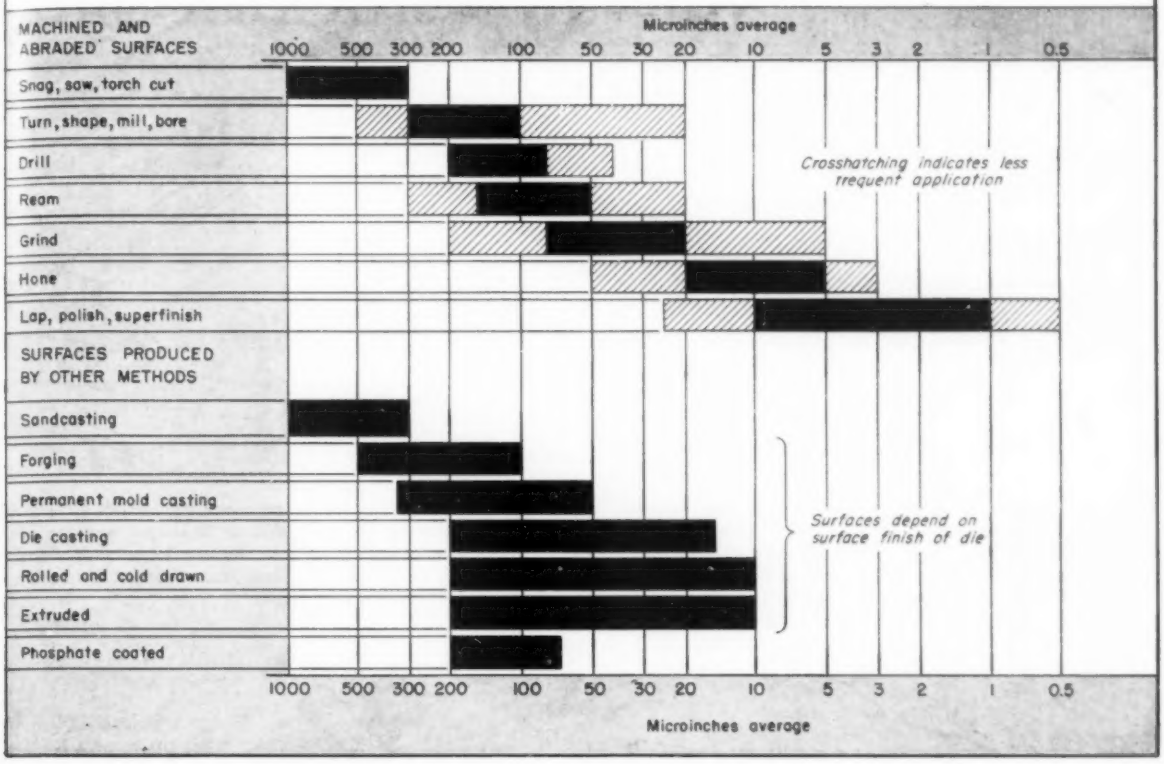
Table I—Surface Finishes Available by Common Production Methods



in the fragments of the built up edge. This helps detect dull tools long before the workpieces must be discarded because of excessive overall roughness or dimensional errors.

A comparison of surface finishes available by common production methods is shown in Table I, reproduced from the surface finish section of the new SAE Drafting Manual. (The major portion of these new surface finish standards are shown on pages 163 to 167, August PRODUCT ENGINEERING). However, the limits of surface roughness obtainable by various machining methods as indicated by the chart are not inflexible. Surface finish is the result of a combination of conditions; changing any one would result in a different surface.

When setting up a processing procedure to obtain a desired surface, the fewest number of steps is usually desirable. However, refining a surface in too large steps may damage the surface. For instance, in grinding operations successive steps should improve the surface roughness by not more than a factor of four.

Non-Machined Surfaces

In this article, the author had machined surfaces in mind for most of the applications cited. What about the odd surfaces that do not fall within the range of these examples? This problem was discussed at an SAE Panel Meeting on Surface Finish (see References). For example, how can one obtain accurate control over castings, shot blasted, grit blasted, and similar surfaces by applying the written surface finish standard in conjunction with present day instrumentation? At present the answer seems to be that these surfaces cannot be controlled merely by the use of symbols and inspection by instruments. The most practical way of controlling these surfaces is to inspect the pieces by comparison with approved specimens. It is likely that for any one concern, control of such surfaces could be fairly effective, but for jobbers and suppliers, perhaps physical specimens or replicas of the desired surface will be required with drawings and instructions.

New instruments and techniques will undoubtedly be developed in the future, and a more general acceptance of present day methods will lead to improved control and product quality.

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Nine Types of Ball Slides

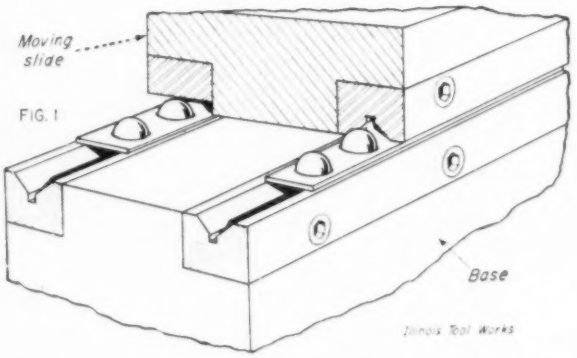


FIG. 1

1 V grooves and flat surface make simple horizontal ball slide for reciprocating motion where no side forces are present and a heavy slide is required to keep balls in continuous contact. Ball cage insures proper spacing of balls; contacting surfaces are hardened and lapped.

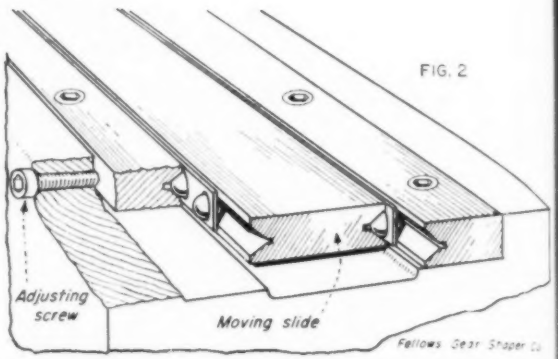


FIG. 2

2 Double V grooves are necessary where slide is in vertical position or when transverse loads are present. Screw adjustment or spring force is required to minimize looseness in the slide. Metal-to-metal contact between the balls and grooves insure accurate motion.

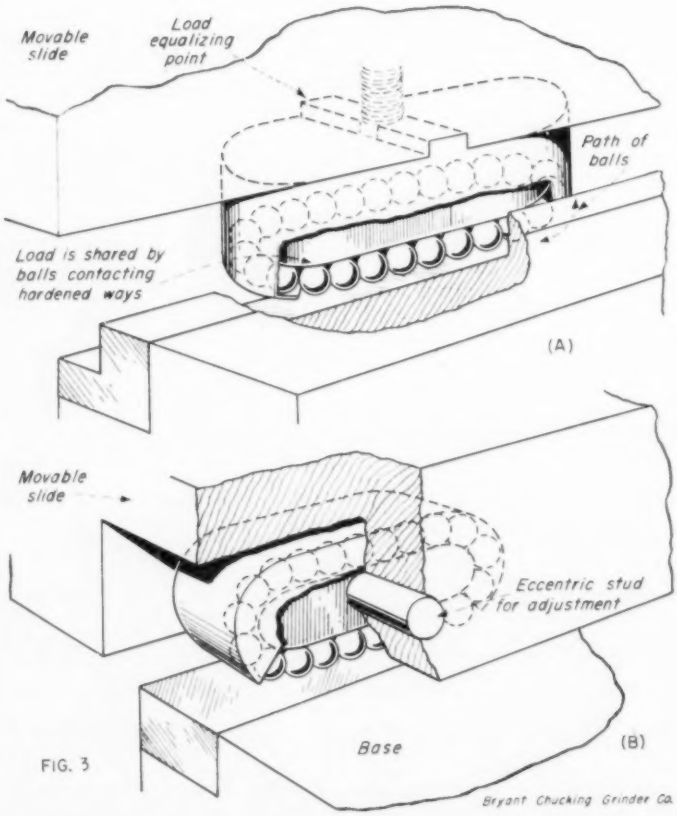


FIG. 3

3 Ball cartridge has advantage of unlimited travel since balls are free to recirculate. Cartridges are best suited for vertical loads. (A) Where lateral restraint is also required, this type is used with a side preload. (B) For flat surfaces cartridge is easily adjusted.

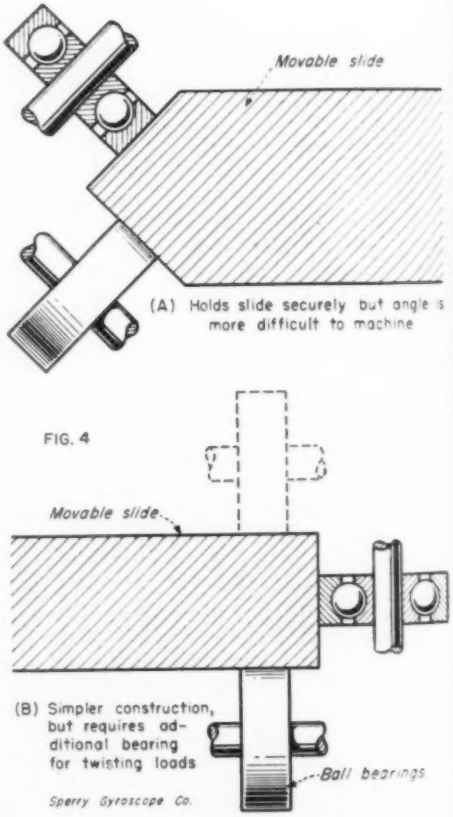
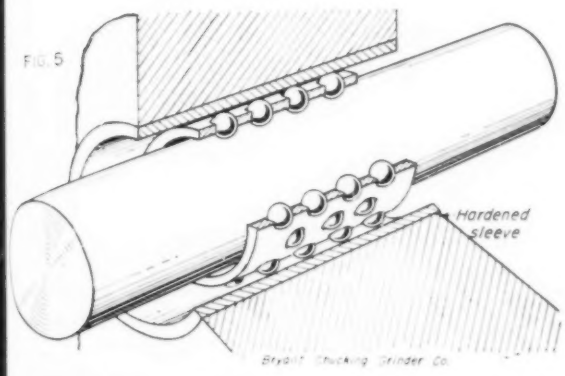

Movable slide
(A) Holds slide securely but angle is more difficult to machine
Movable slide
(B) Simpler construction, but requires additional bearing for twisting loads
Ball bearings
Sperry Gyroscope Co.

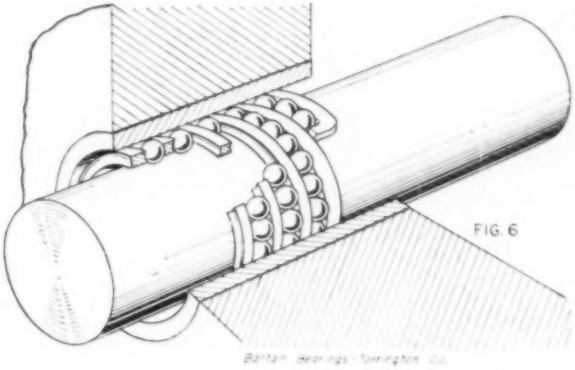
FIG. 4

4 Commercial ball bearings can be used to make a reciprocating slide. Adjustments are necessary to prevent looseness of the slide. (A) Slide with beveled ends, (B) Rectangular-shaped slide.

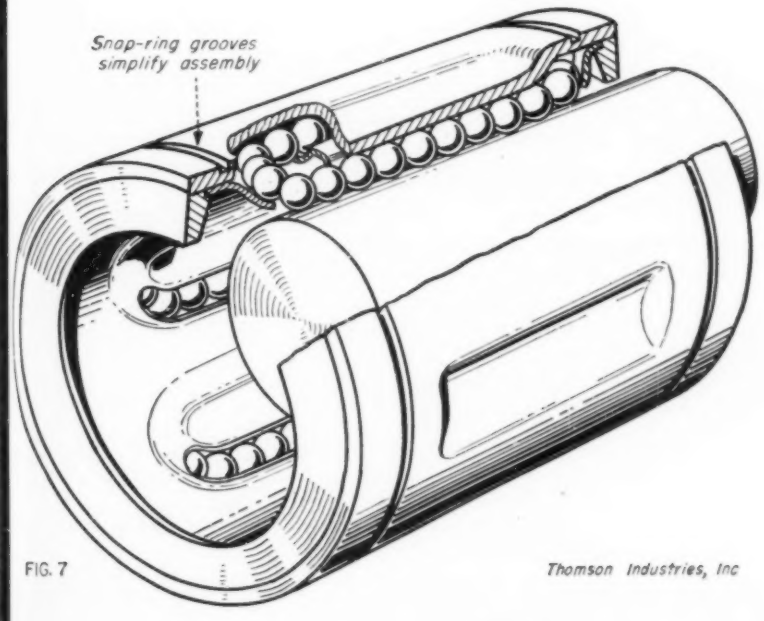
For Linear Motion



5 Sleeve bearing consisting of a hardened sleeve, balls and retainer, can be used for reciprocating as well as oscillating motion. Travel is limited similar to that of Fig. 6. This type can withstand transverse loads in any direction.



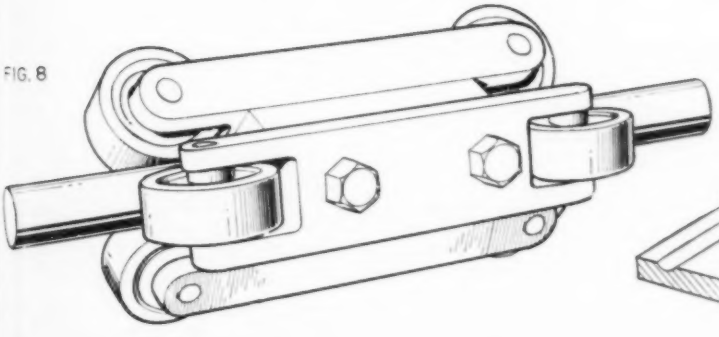
6 Ball reciprocating bearing is designed for rotating, reciprocating or oscillating motion. Formed-wire retainer holds balls in a helical path. Stroke is about equal to twice the difference between outer sleeve and retainer length.

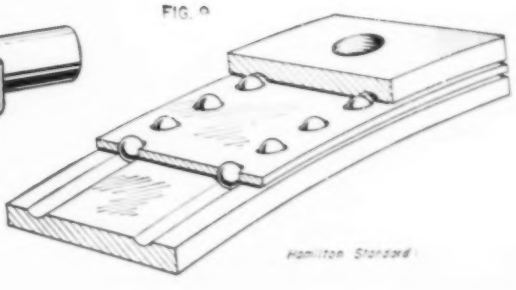


7 Ball bushing with several recirculating systems of balls permit unlimited linear travel. Very compact, this bushing simply requires a bored hole for installation. For maximum load capacity a hardened shaft should be used.

8 Cylindrical shafts can be held by commercial ball bearings which are assembled to make a guide. These bearings must be held tightly against shaft to prevent looseness.

9 Curvilinear motion in a plane is possible with this device when the radius of curvature is large. However, uniform spacing between grooves is important. Circular - sectioned grooves decrease contact stresses.





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Aluminum Die Forging Design Details That Promote Metal Flow*

Factors that influence the dimensions of corner radii at section edges and of fillets at intersections and junctions of ribs and webs. Design of punch out holes in thin webs and ribs. Design proportions for forged pockets and recesses.

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Forging Division, Aluminum Company of America

IN ADDITION TO OBSERVING the design rules for side draft, bottom draft, and the location of parting lines as discussed in Part I of this article, PRODUCT ENGINEERING, August 1950, p 140, designers can promote the efficient production of die forged aluminum alloy parts by using generous fillet and corner radii.

Forming a part with abrupt changes in section from a cylindrical or rectangular bar deforms the forging stock severely. The number of operations needed to bring the piece to its final shape depends entirely on the design details of the forging.

Liberal fillets on forgings, as shown in Fig. 1, allow the forging stock to follow the die contour while being deformed. If a fillet is not large enough, the relatively sharp corner on the die may cause a momentary void to form beyond the bend in the flow path. When the die is filled, the void is folded shut, but it leaves a flaw in the forging. Such flaws, called "cold shuts", do not weld together in the aluminum and magnesium alloys, and finished forgings that have such defects must be rejected.

Adequate fillets between sections minimize uneven shrinkage as the metal cools after forging. They also minimize distortion later during the quenching of heat-treatment.

Maintain a constant fillet radius on forgings to keep die cost low. The die sinker spends extra time blending rounded edges of different radii on dies, and such work should be avoided when it is not necessary.

There are designs where blending of fillets on a forging is necessary. Fig. 2 shows a typical example in which a heavy end rib requires a large fillet, while a smaller fillet is used along the adjacent tapered rib to provide a maximum flat surface on the web. The two radii are blended smoothly throughout the curved intersection. This operation is done as a routine matter in die sinking without the need for an explanatory note on the drawing. For guidance of designers, minimum fillet radii are specified in Fig. 3. Adherence to these standards is usually essential to the production of sound forgings, and any radical

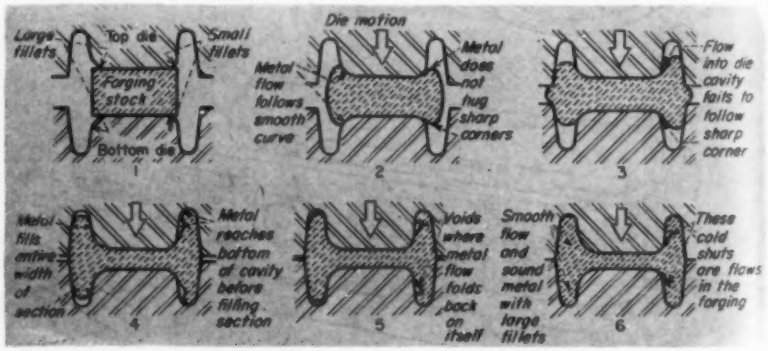


Fig. 1—To emphasize the effect of fillet radius on metal flow, this forged section is shown with large fillets on the left side and small fillets on the right side. The steps demonstrate how small fillets may cause unfilled sections, laps and cold shuts.

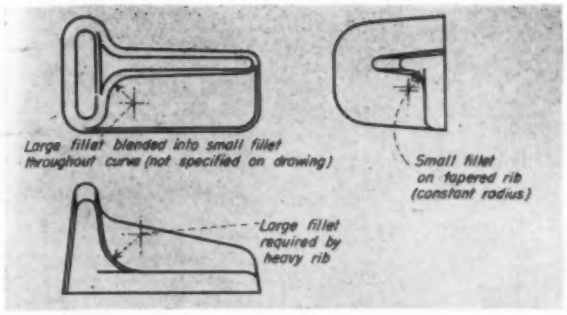

Large fillet blended into small fillet throughout curve (not specified on drawing)
Small fillet on tapered rib (constant radius)
Large fillet required by heavy rib

Fig. 2—When necessary to blend fillets of different radii, make the change of radius at a corner between uniform surfaces.

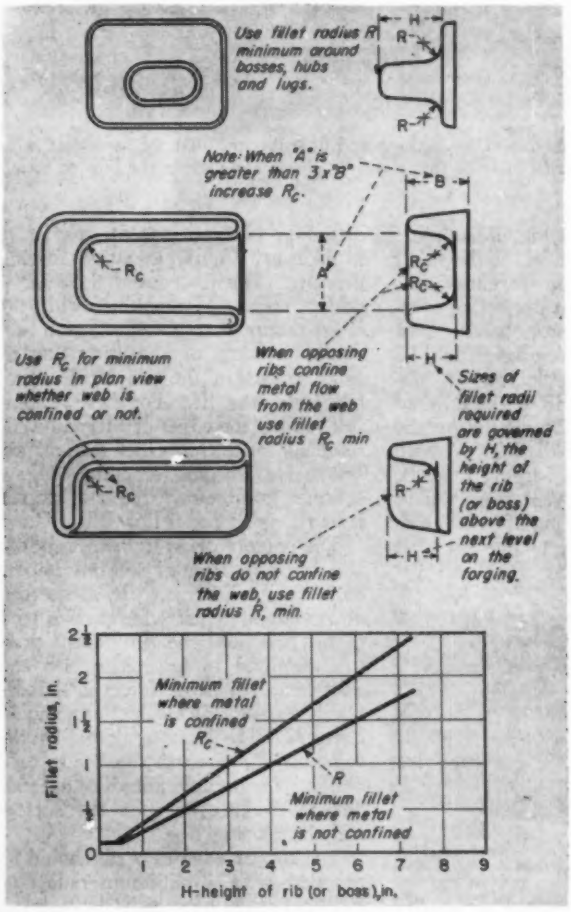

Use fillet radius R minimum around bosses, hubs and lugs.
Note: When "A" is greater than 3 x "B" increase R_c.
Use R_c for minimum radius in plan view whether web is confined or not.
When opposing ribs confine metal flow from the web use fillet radius R_c min
Sizes of fillet radii required are governed by H, the height of the rib (or boss) above the next level on the forging.
When opposing ribs do not confine the web, use fillet radius R, min.
Minimum fillet where metal is confined
Minimum fillet where metal is not confined
Fillet radius, in.
H-height of rib (or boss), in.

Fig. 3—Recommended minimum fillet radii on ribs and bosses in confined and unconfined metal of die forged aluminum parts.

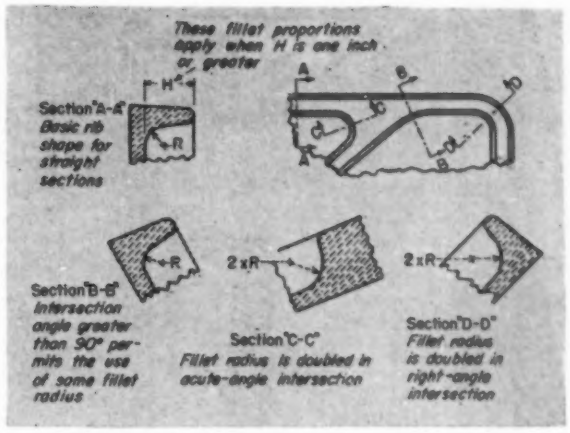

These fillet proportions apply when H is one inch or greater
Section "A-A" Basic rib shape for straight sections
Section "B-B" Intersection angle greater than 90° permits the use of some fillet radius
Section "C-C" Fillet radius is doubled in acute-angle intersection
Section "D-D" Fillet radius is doubled in right-angle intersection

Fig. 4—Where high ribs intersect at angles of 90 deg or less, rib-to-web fillets have double the basic radius. Increasing these radii is required to prevent cold shuts at the rib intersections.

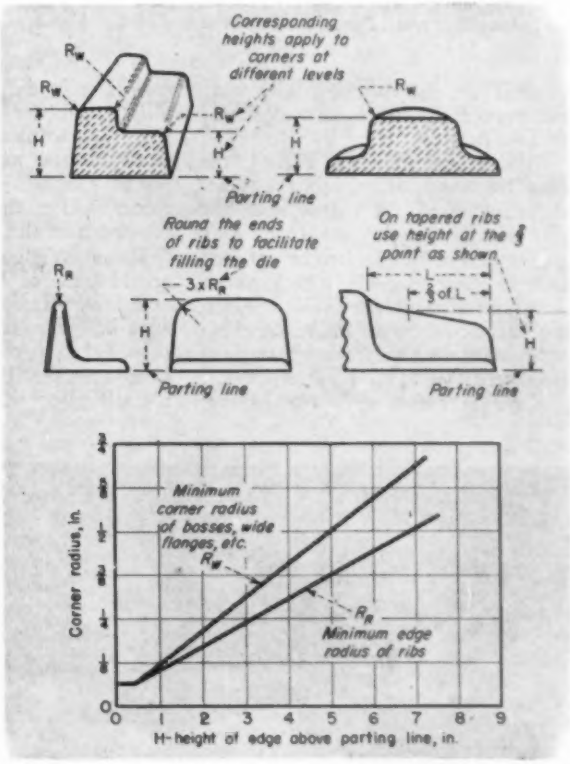

Corresponding heights apply to corners at different levels
Parting line
Round the ends of ribs to facilitate filling the die
On tapered ribs use height at the ⅔ point as shown
Minimum corner radius of bosses, wide flanges, etc.
Minimum edge radius of ribs
Corner radius, in.
H-height of edge above parting line, in.

Fig. 5—Recommended minimum radii at corners and edges for assuring good metal flow and to obtain satisfactory die life.

exceptions made only after study by a competent forging supplier.

Where the flow of metal to the gutter at the parting line is confined, use a radius equal to R_c (minimum) for the rib-to-web fillets. If the forging stock must fill in ribs on opposite sides of the web before flowing out of the impression as a flash, fillets at least as large as R_c are needed to assure sound metal structure. They prevent "flow through" with consequent rupture of the grain flow pattern, and they reduce likelihood of cold shuts.

The minimum values of the R_c given in Fig. 3 are for forgings of average proportions. If the web is very broad and the ribs are relatively shallow, larger fillets may be advisable. The recommendations of the forging producer's engineers should be followed when the width of the web exceeds three times forging height.

On the plan view of a box-shaped forging, a fillet between adjoining ribs will have a varying radius because of draft on the sides of the ribs. As indicated on the web at the base of the ribs, the smallest value of this radius should equal or exceed R_c. Where the flow of metal from the web to the gutter is not confined by ribs on opposite sides, smaller fillets can be used at the bases of the ribs. Values given for R in Fig. 3 are satisfactory.

Bosses and lugs, even though inside

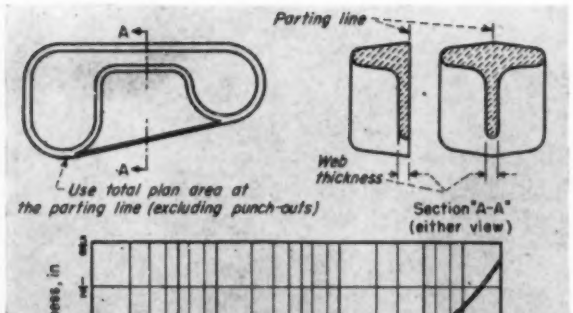


Fig. 6—Minimum web thickness of aluminum die forgings based on parting line areas. Punch out holes not included in plan area.

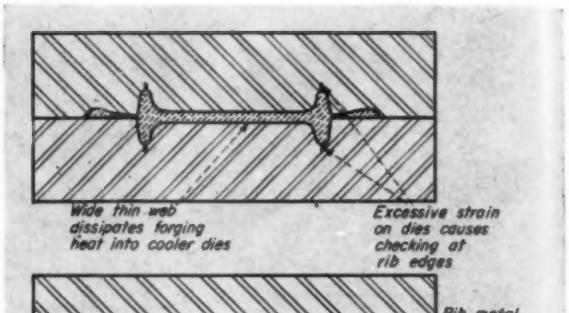

Wide thin web dissipates forging heat into cooler dies
Excessive strain on dies causes checking at rib edges
Rib metal cooled too soon
Gutter
Rib section sheared off at base by flow of web metal to gutter
Defect known as "flow through"

Fig. 7—Die checking and flow through that are associated with ribs or wide thin webs. Thin sections impede the flow of metal.

channel or dish sections, may have minimum fillet radii equal to R, based on boss height.

Where ribs intersect, Fig. 4, the basic rib-to-web fillets radius is sometimes increased. A corner with the ribs at right angles or at an acute angle, and with rib heights of one inch or more, is made with a rib-to-web fillet of twice the radius used on the straight portions. Increasing these radii is required to prevent cold shuts from occurring at the rib intersections.

Constant corner radii on forgings save time in die sinking. It is more difficult to blend radii in the die fillets that form corners on forgings than to do the same job on projecting corners of the die. Hence, blending of corner radii on the forgings should be avoided if at all possible.

Minimum edge radii are established on the basis of facility in filling die cavity and reasonable die life. A sharp edge or corner on a forging must be shaped by a correspondingly sharp fillet in the die. Forcing the metal into a small fillet, which is the last part of the die impression to fill, may make it necessary to use excessive forging pressure. The force exerted by metal under pressure, combined with the stress-raising corner effect, can develop checking or the formation of small cracks in the die after a prohibitively short life. For these reasons, full radii at the edges of ribs and well-rounded corners on other sections are essential to forging economy. Fig. 5 specifies minimum edge and corner radii for good metal flow.

The corners of wide flanges, bosses and large rectangular sections impose more severe loads on die fillets than do rib edges. Their wide cavities tend to fill first, and the subsequent pressure required to fill narrow rib sections creates excessive forces in already-filled cavities. Wide sections should have radii at least as large as the values of R_w given in Fig. 5. In each design, the minimum radius depends on the distance between the surface and the parting line.

The top or edge of a rib should be rounded with a minimum radius of R_R, the value of which depends on rib height measured from the parting line. The end of a rib should be rounded to avoid a deep corner pocket that would be difficult to fill. This end taper should have a radius equal to at least three times the edge radius.

Minimum edge radii limit the use of excessively thin high ribs. With a full radius at its edge and a standard draft on the sides, a rib cannot be thinner than twice R_R. A thicker rib may have a flat edge with two corner radii, each equal to R_R. A full radius equal to half the rib thickness is much more desirable.

Thin sections of large area cool

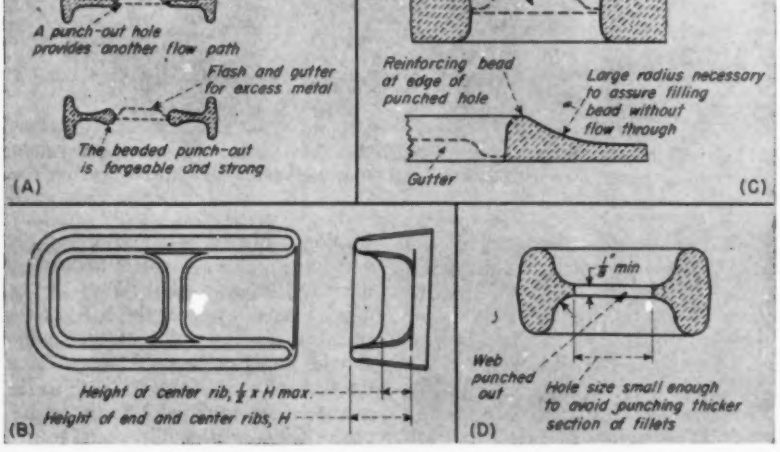

4° or more
A tapered web aids metal flow to ribs
Flash and gutter for excess metal
A punch-out hole provides another flow path
Flash and gutter for excess metal
The beaded punch-out is forgeable and strong
(A)
Excess metal in flash and gutter punched out after finish forging
Reinforcing bead at edge of punched hole
Large radius necessary to assure filling bead without flow through
Gutter
(C)
Height of center rib, ½ x H max.
Height of end and center ribs, H
(B)
⅛" min
Web punched out
Hole size small enough to avoid punching thicker section of fillets
(D)

Fig. 8—Designs for forgings that must have thin webs and ribs. (A) Use of tapered webs and punch-out holes. (B) Proportions for center, side and end ribs. (C) Punched holes used to reduce weight and to aid production. (D) Hub forging with punch-out.

rapidly in the forging dies. They need repeated blows or high pressures to forge them down to size. This cooling and this operation can impose excessive strain on dies and equipment, and may cause other difficulties.

Dimensional tolerances of a thin web are hard to maintain because the web cools and shrinks faster than other portions of the forging. While the remainder of the forging is cooling, normal shrinkage is partially restrained by the web, and the piece comes out with abnormal dimensional relationships. Thin sections are also susceptible to warping in heat-treatment. They may have to be straightened, which is difficult and expensive.

Minimum web thickness, in general, depends on the size of the forging expressed as area on the parting line. Fig. 6 gives recommended limits for thin webs. Note that punch-out holes are not included in plan area when minimum web thickness is determined.

Die life is reduced when deep, thin ribs are forged. The force that is needed to fill a deep rib cavity, especially when it is adjacent to a very thin web, results in severe die stresses. For this reason, ribs that are next to very thin webs should be made as wide as possible and as low as the design will permit.

Flow through is a type of forging defect that may be caused by insufficient rib and web thickness. Cooling of the metal during reduction of thin sections increases the metal flow resistance. As shown in Fig. 7 the rib metal may cool (with or without filling the section) before the web is forged down to size. Then, further reduction of the web forces excess metal past the bases of the ribs into the gutter, which is the path of least resistance. Each rib has an area of ruptured metal at the base, and the forging cannot be used.

Helpful design methods for forgings that must have thin webs and ribs are shown in Fig. 8. A tapered web or a punch-out in the web will make the forging problem less severe.

Center ribs, which may be needed for strength, should be less than half the height of side and end ribs. Fig. 8(B) shows the type of rib, surrounded by other ribs, that is covered by this rule. It is possible to forge center ribs more than half the height of the surrounding bosses and ribs, but such a design invites considerable forging difficulty.

A punch-out is a sheared hole in a forged web. The web is preformed with a depression or pocket, leaving only a thin panel to be punched out in a separate operation.

Thin webs confined by ribs are much easier to forge if provided with

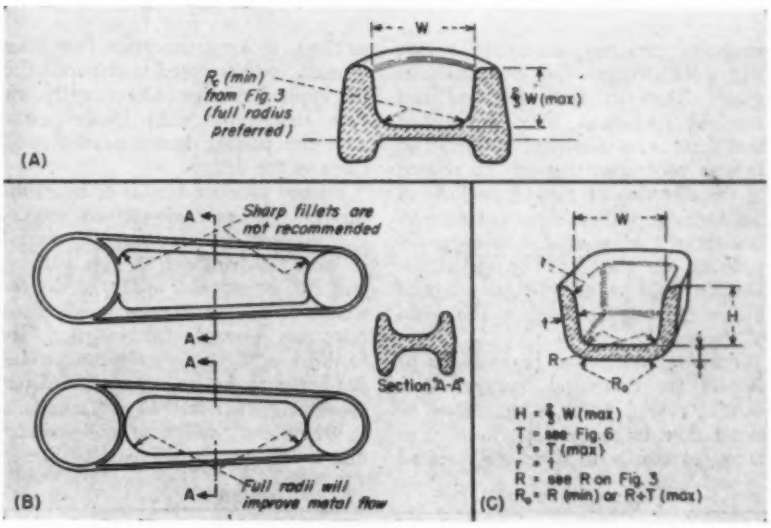


Fig. 9—(A) Proportions for forged pockets. (B) Keep recesses as smooth and simple as possible. (C) Proportions of bathtub type forgings when parting is at rib edges.

punch-out holes. Punch-outs are also highly recommended on very large forgings. It is customary to pierce the hole first, during the preforming stages of making the forging. Then, in the finishing dies there is gutter for flash around the periphery of each hole. With this arrangement, excess metal from the web can escape easily. Additional paths for excess metal minimize the harmful tendency to flow through at the bases of ribs, and they facilitate working the forging down to size. The flash in the hole may be trimmed along with the flash left on the outer parting line.

Punched holes in webs and flanges are used to reduce weight or to provide clearance as well as to aid production Fig. 8(C). Where very high strength is desired, such a hole can be reinforced with a bead like the one shown. The bead should be blended into the web with a large fillet to prevent flow through, which has a tendency to occur at these locations.

Hubs and bosses may have punch-outs to facilitate boring and other machining that require through holes, Fig. 8(D). The punch hole will leave a ridge at the fillets; this ridge must be removed by machining if its presence is undesirable. Generally, these punched holes detract from rather than add to the ease of forging.

The proportions of forged pockets that govern the depth of forged recesses left for piercing are given in Fig. 9(A). If the part is so simple that only a finishing impression is necessary, the web to be punched will need generous fillets where it joins the boss.

Recesses in forged parts are formed by corresponding raised sections or "plugs" in the dies. In the interest of die life and reliability, a forged pocket should not exceed two-thirds of its diameter or two-thirds of its average width. Simple contours and generous fillets are recommended, Fig. 9(B), because they permit the use of massive, durable die shapes, and they also promote metal flow.

Dished or bathtub type forgings, like the one shown in Fig. 9(C), are impractical when the wall thicknesses become too heavy. The walls should be no thicker than the web, which can be forged extra-heavy if desired and later machined down.

Aluminum forgings gain a substantial increase in toughness and reliability through improvement of grain structure in the metal. When forging design and die design are properly worked out, the fibrous metal "grain" should be lined up parallel to the most severe stresses and to the shock loads that will be imposed on the part.

The forging stock in bar or rod form has a directional structure imparted to the cast ingot metal by a pressing, rolling or extruding operation. During forging, while the metal is in the plastic state, this fibrous grain structure is deformed. It is made to flow under pressure into the shape of the forging.

By taking advantage of metal flow, maximum physical properties can be obtained at critically stressed sections. Such a result is not possible with any of the common metal-working processes except forging.

Tensile and yield strengths measured parallel to grain flow are always maximum for a given forging; in most

forgings, however, transverse tensile and yield strengths are practically as great. Tests of both polished and notched specimens have established that there is no consistent difference in fatigue properties that can be related to the direction of metal flow lines in the forging. Other important properties show a substantial difference, depending on the direction of stress. Ductility and impact strength are both higher measured parallel to the grain of the metal.

Forgings subject to heavy loads or impact are therefore designed with careful regard for the directions of metal flow in stressed areas. Care is taken, especially at fillets and reduced sections, to keep the grain flow lines smooth, uninterrupted, and parallel to the applied stresses. Occasionally, the flow characteristics of forged metal near the parting line assume importance in the design.

Forged pressure vessels or hydraulic fittings, for example, often require special attention. If there are several cylindrical chambers in the part, it will not be possible to forge the interiors of all the recesses in the same way that pistons, for instance, are handled in a mechanical upsetter. Instead, the chambers are machined out of solid cylindrical forged sections.

When the portion of the cylinder that is adjacent to the parting line is machined to a thin wall, the only metal left in this area has the sharply interrupted grain structure created by flow out into the gutter. A tendency to develop seepage or small pin hole leaks under high pressure may result from this condition.

The remedy is to leave a bead along the edge of the cylinder at the parting line. The bead may be machined off later to improve appearance or to save weight, but the bead will have served its purpose by removing the parting line irregularities a short distance away from the cylinder wall proper. The grain of the metal behind the bead is transverse to the stressed direction, but it is smooth and sound.

Two-Tooth Gear Systems

S. RAPPAPORT
Ford Instrument Company

THE PROFILE OF A GEAR TOOTH is determined by the laws of kinematics under the assumption that two coplanar pitch circles roll, without sliding, around each other in opposite directions at equal circumferential speed. But what happens to the gear system if this assumption is modified by the condition that the two planes must rotate at equal speed in the same direction?

Applying the laws of relative motion in the diagram, Fig. 1, the point P is located at distance r from the rotating center A of one plane. Relative to the second plane, which rotates around center B, point P describes a circle having the radius $r\sqrt{2}$.

But the two gears generated using radius $r\sqrt{2}$ do not make up a constrained kinematic chain because the movement of one gear does not completely control the movement of the other. To make one gear precisely define the movement of the other at all times, each gear must consist of a set of at least three two-tooth gear laminations. The latter are spaced at equal angles around the gear center as shown in Fig. 2.

Four of these laminated two-tooth gears mounted in mesh with their centers at the corners of a square fulfill the original condition as stated above. If one of the four gears is rotated, the other three will turn in the same direction. Similarly, five or more of these gears arranged, in mesh, around a circle would increase the number of centers.

Although there is an inherent high friction loss due to the constant sliding of the tooth tip of one gear along the face of the other, which results in very low coefficiency of the gear system, there are undoubtedly possible applications for this type of gear mechanism. It is not suitable for use as a power transmission element, but the pocket formed between the teeth as they rotate could conceivably be utilized for metering semi-solid materials.

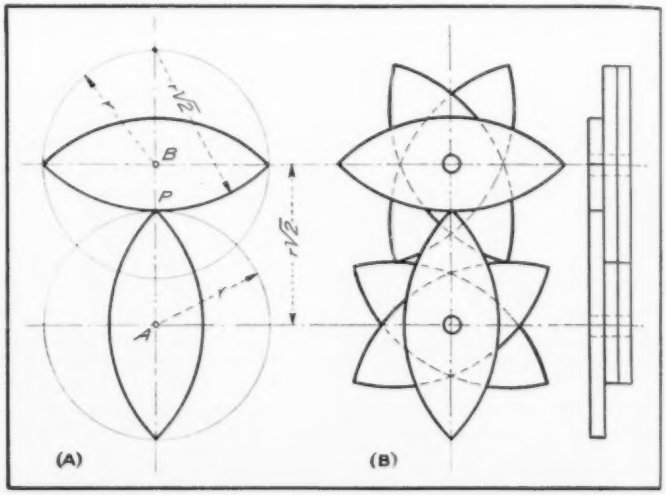
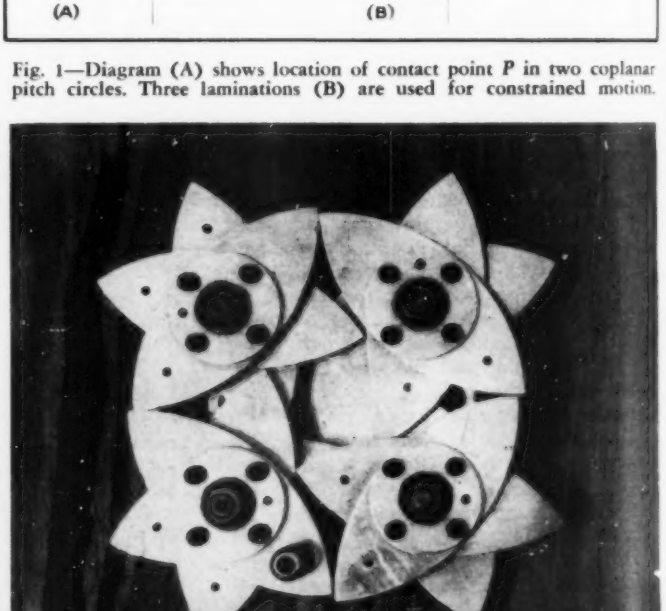


Fig. 1—Diagram (A) shows location of contact point P in two coplanar pitch circles. Three laminations (B) are used for constrained motion.



Wright Machinery Co.

Fig. 2—Four or more gears equally spaced on a circle have same rotation.

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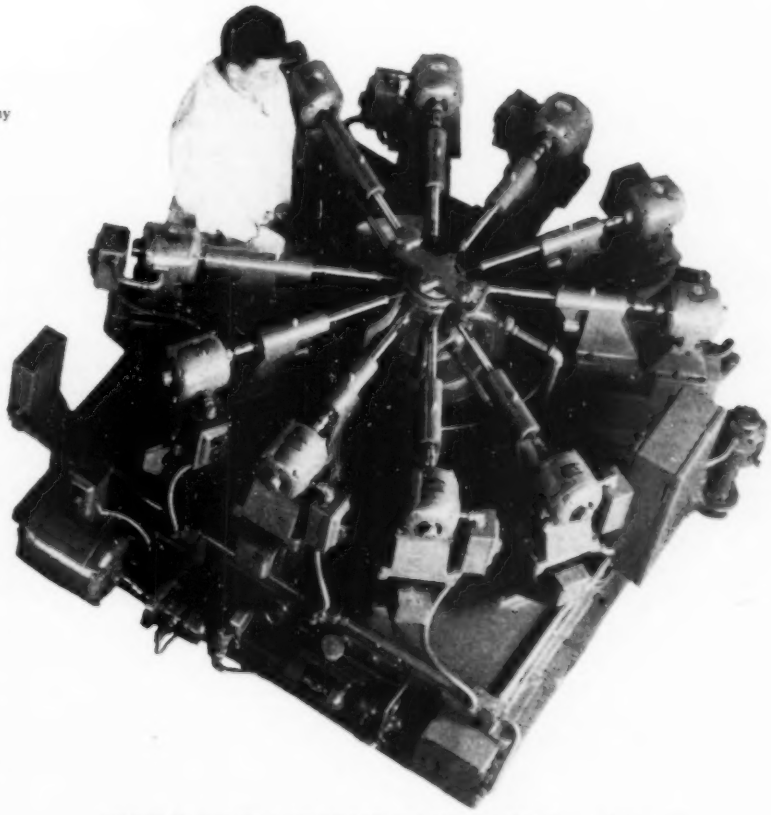
PART II

Determination Of

Motor Horsepower From Duty Cycle

L. H. BERKLEY

Application Engineer, The Master Electric Company



TYPICAL SPECIAL PURPOSE MACHINE TOOL employing a number of motors in a coordinated operation. In many such cases the duty cycle of each motor must be analyzed individually.

IN a previous article (PRODUCT ENGINEERING, Aug. 1950) duty cycles for varying loads and frequent acceleration were discussed. A third type of severe duty cycle is one where the drive motor must plug-reverse. The motor actually does the braking in addition to the usual functions of accelerating and carrying the load.

In plug-reversing applications also motor rating can be calculated accurately if the duty cycle of the equipment and the torque-speed characteristics of the motor are known. From this information, the design engineer can determine the equivalent horsepower of each part of the duty cycle. The root-mean-square of these equivalent horsepower values is the motor rating required to drive the equipment without overheating.

The formula for equivalent horsepower during acceleration was established in the previous article as:

$$HP_e = \frac{T_{avg} \times RPM}{5{,}252} \qquad (4)$$

Where T_{avg} = average torque lb-ft delivered during acceleration.

RPM = rated motor speed, rpm.

The heating during a plug-reversal of an induction motor is somewhat more severe than that during acceleration. Accordingly, Formula 4 is modified by a multiplying factor of $\sqrt{1.5}$ (or 1.225) in calculating equivalent horsepower for plug-reversals. This indicates that the average I^2R loss during plug-reversal is 1.5 that during acceleration. The factor, which works out well in practice, is based on several assumptions:

1. During acceleration, the rotor and stator I^2Rt losses (watt-seconds) are equal. Actually, the relative magnitudes of the two vary somewhat with the design (B, C or D) of the motor.

2. RMS stator current during plug-reversal equals that during acceleration. Current during plug-reversal is actually higher because of the higher current during the plug-to-rest period.

3. Rotor I^2Rt losses during plug-reversal is four times that during acceleration. This can be proved mathematically.

4. The duration of a plug-reversal is double that of an acceleration. Actually, the plug-to-rest period is somewhat shorter than the acceleration period. It will be noted that the inaccuracies of assumptions 2 and 4 tend to cancel one another.

Considering the total I^2Rt loss during acceleration as unity, stator and rotor losses are each about equal to one-half. During plug reversal, the stator I^2Rt loss is twice that during acceleration, or a total of one. Also, rotor loss is four times that during acceleration, or two. Rotor and stator

losses total three, indicating that the I^2Rt loss during a plug-reversal is three times the I^2Rt loss during acceleration. Since a plug-reversal lasts twice as long as an acceleration, the rate of loss (I^2R) is 1.5 times the I^2R during acceleration. Because equivalent horsepower varies as the square root of the losses, the equivalent horsepower during plug-reversal is $\sqrt{1.5}$ that during acceleration.

Therefore, during plug reversal of Design B motors:

$$HP_e = \sqrt{1.5}\,\frac{T_{avg} \times RPM}{5{,}252}$$

$$= \frac{T_{avg} \times RPM}{4{,}290} \quad (12)$$

$$= 1.225\,\frac{T_{avg}}{T_{rated}}\,HP_{rated} \quad (13)$$

And, during plug-reversal of Design-D motors:

$$HP_e = \sqrt{0.75 \times 1.5}\,\frac{T_{avg} \times RPM}{5{,}252}$$

$$= \frac{T_{avg} \times RPM}{4{,}950} \quad (14)$$

$$= 1.06\,\frac{T_{avg}}{T_{rated}}\,HP_{rated} \quad (15)$$

Except for these formula differences, the analysis of a plug-reversing duty cycle is similar to one involving frequent acceleration.

Reasoning similar to that used above for plug-reversal is also applicable to the plug-stopping of induction motors. It can be determined that the total I^2R loss when a motor is plugged to a stop is about 3.5 times the I^2R loss during acceleration. Therefore, during a plug-stop of a Design-B motor:

$$HP_e = \sqrt{3.5}\,\frac{T_{avg} \times RPM}{5{,}252}$$

$$= \frac{T_{avg} \times RPM}{2{,}810} \quad (16)$$

$$= 1.87\,\frac{T_{avg}}{T_{rated}}\,HP_{rated} \quad (17)$$

And, for Design-D motors:

$$HP_e = \frac{T_{avg} \times RPM}{3{,}240} \quad (18)$$

$$= 1.62\,\frac{T_{avg}}{T_{rated}}\,HP_{rated} \quad (19)$$

The basis for calculating equivalent horsepower of Design-D motors during acceleraton and plug-reversal was given in the first part of this article. In applying the formulas, it should be borne in mind that there is no definite dividing line between Design-B and Design-D motors. There are many transition designs used for certain applications.

Plug-Reversing Application

A medium-size lathe, used for high production machining of small metal fittings, was powered originally by a two hp polyphase gearmotor. The motor operated at 1,725 rpm while 518 rpm was the countershaft speed. The countershaft was belt connected to the lathe spindle. The high production rate of 180 pieces per hour was found to cause severe overheating of the motor. The problem was to select the proper motor rating.

The torque-speed characteristic of the Design-B motor is given in Fig. 7, while Fig. 6 shows the equivalent-horsepower duty cycle. Note that the motor is never idle. The fitting to be machined is manually threaded onto the spindle. After it is machined, the motor is plug-reversed and the fitting removed in similar fashion. The motor is then again plug reversed and another fitting inserted.

Calculation of the rms horsepower follows the same pattern as for frequent starting. The equivalent horsepower during plug-reversal is:

$$HP_e = \frac{14.4 \times 1{,}725}{4{,}290}$$

$$= 5.79$$

From knowledge of the gear characteristics and the machine, the average friction load at the motor shaft during acceleration is estimated at 1.5 lb-feet. Hence, the torque available for acceleration (T_{acc}) is 12.9 lb-feet. From the weights and dimensions of the pulleys, the total effective inertia at the motor shaft was calculated to be 1.28 lb-feet^2. The plug-reversing time, which is approximately twice the accelerating time, is therefore:

$$t_{p-r} = 2\,\frac{WR^2 \times S}{308\,T_{acc}}$$

$$= 2\,\frac{1.28 \times 1{,}725}{308 \times 12.9}$$

$$= 1.11 \text{ sec}$$

The equivalent running friction of the machine at the motor shaft is 1.25 lb-ft or 0.4 horsepower. This is also used as the horsepower during the filing operation since the additional load caused by filing is negligible. From tests during the peak load, it was determined that the motor was running at 250 percent load (5.0 hp) and at 1,500 rpm approximately. The equivalent horsepower during the overload period is then:

$$HP_e = \text{Actual Peak HP} \times \frac{\text{Speed at Full Load}}{\text{Speed at Overload}}$$

$$= 5.0 \times \frac{1{,}725}{1{,}500}$$

$$= 5.75 \text{ hp}$$

Using the above data, the equivalent horsepower duty cycle of Fig. 6

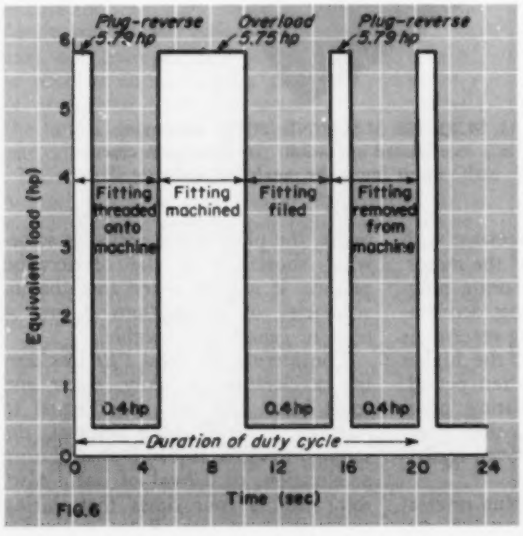


FIG. 6

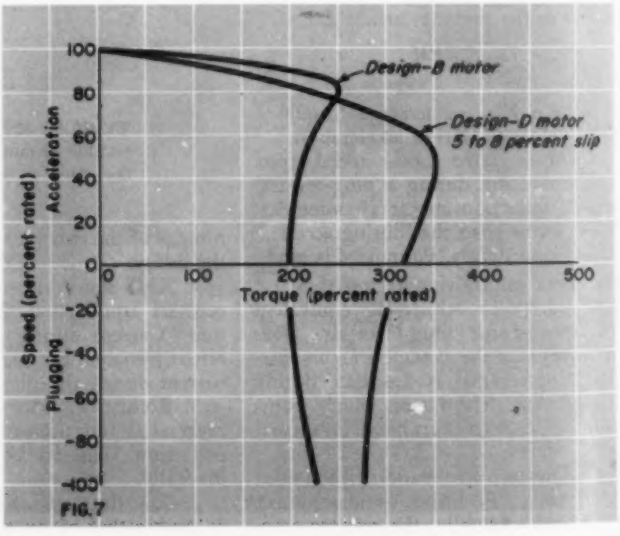


FIG. 7

is established. The rms horsepower is calculated as follows:

$$HP_{rms} = \sqrt{\frac{(HP_1)^2 t_1 + (HP_2)^2 t_2 \ldots (HP_n)^2 t_n}{t}}$$

$$HP_{rms} = \sqrt{\frac{(74.4) + (12.2) + (166)}{0.5\,(1.11 + 1.11) + (17.78)}}$$

The first term of the numerator represents the two plug-reversals.

The second term, the periods of light load; and the third term; the period of overload.

The first term of the denominator represents the plug-reversing time and the second term, the remainder of the duty cycle when the motor is running at or near rated speed.

$$HP_{rms} = \sqrt{\frac{74.5 + 2.1 + 165}{18.89}} = 3.58$$

This load represents a continuous overload of 80 percent on a two hp motor, which is obviously insufficient for the duty cycle. Note that the heating due to the steady running period is 2.2 times the heating during the accelerating period. Consequently, little reduction in rms horsepower can be effected by using a Design-D motor, even if it reduced the heating during acceleration as much as 25 percent. Furthermore, a Design-D motor is undesirable because the high slip during overload reduces production.

A three hp Design-B motor may be satisfactory. Such a motor increases the relative heating during acceleration (because its WR^2 is higher) and reduces the equivalent overload (because it will operate at a higher speed). To ascertain whether or not a three

Customer: XYZ FITTINGS, INC.

Machine: PRODUCTION LATHE

General Information (WR^2, loads, duration, etc.):

DUTY CYCLE - SEE FIG

TOTAL EFFECTIVE WR^2 = 1.28 #-ft.^{2}

Sequence (a)	Part of Cycle	Equiv. Torque Note	Equiv. Torque Ft.Lbs.	Equiv. HP Note	Equiv. HP HP	Actual Time Note	Actual Time Sec	$HP^2 t$	Eq.Time K	Eq.Time S
	Acceleration	(c)		(d)		(e)			(b)	
	Running	(f)		(f)(g)		(f)			(b)	
	Plug-Reverse	(c)		(h)		(i)			(b)	
	Plug-Stop	(c)		(j)		(e)			(b)	
1	PLUG-REVERSE	(c)	14.4	(12)	5.79	(8a)	1.11	37.2	.50	.56
2	FITTING ATTACHED	(f)	1.25	(4a)	0.4	(f)	3.89	0.6	1.03	3.89
3	" MACHINED	(f)	—	(1)	5.75	(f)	5.0	165.3	1.05	5.0
4	" FILED	(f)	1.25	(4a)	0.4	(f)	5.0	0.8	1.05	5.0
5	PLUG-REVERSE	(c)	14.4	(12)	5.79	(8a)	1.11	37.2	.50	.56
6	FITTING REMOVED	(f)	1.25	(4a)	0.4	(f)	3.89	0.6	1.03	3.89
7										
8										
9										
10										
11										
12										
13										
14										
						Totals →		241.7		18.9

$$\text{RMS HP} = \sqrt{\frac{\text{Total } HP^2 t}{\text{Total Equiv. Time}}} = \sqrt{\frac{241.7}{18.9}} = 3.58 \text{ HP}$$

(a) Time sequence - 1st (1), 2nd (2), etc.

(b) Table of Constants (K - Equiv.Time):

DESCRIPTION OF MOTOR	RUNNING FULL SPEED	ACCELERATING OR BRAKING	REST
Self-ventilated up to 10 HP	1.0	0.5	0.25
Self-ventilated above 10 HP	1.0	0.67	0.33
Forced-ventilated all ratings	1.0	1.0	1.0

(c) See Speed-torque curve.

(d) For Design-A or -B: $HP_e = \frac{T_{avg} \times RPM}{5252}$ (4)

$$HP_e = \frac{T_{avg} \times RPM}{6070} \quad (10)$$

(e) $t_{acc} = \frac{WR^2 S}{308\ T_{ace}}$ (8)

(f) Such information as running torque or horsepower, duration of running period, etc. must be furnished by user or established by observation or test.

(g) At overload, $HP_e = HP_{act} \frac{\text{Speed at full load}}{\text{Speed at overload}}$

(h) For Design-A or -B: $HP_e = \frac{T_{avg} \times RPM}{6290}$ (11)

For Design-D: $HP_e = \frac{T_{Avg} \times RPM}{4060}$ (14)

(i) $t_{p-r} = \frac{WR^2 S}{154\ T_{ace}}$ (8a)

(j) For Design-A or -B: $HP_e = \frac{T_{avg} \times RPM}{7010}$ (16)

For Design-D: $HP_e = \frac{T_{avg} \times RPM}{5710}$ (18)

WORK SHEET for equivalent horsepower analysis of duty cycle of the lathe. All required formulas are included. Sheets such as this are invaluable to the application engineer seeking a rapid, yet accurate, guide for duty cycle analysis.

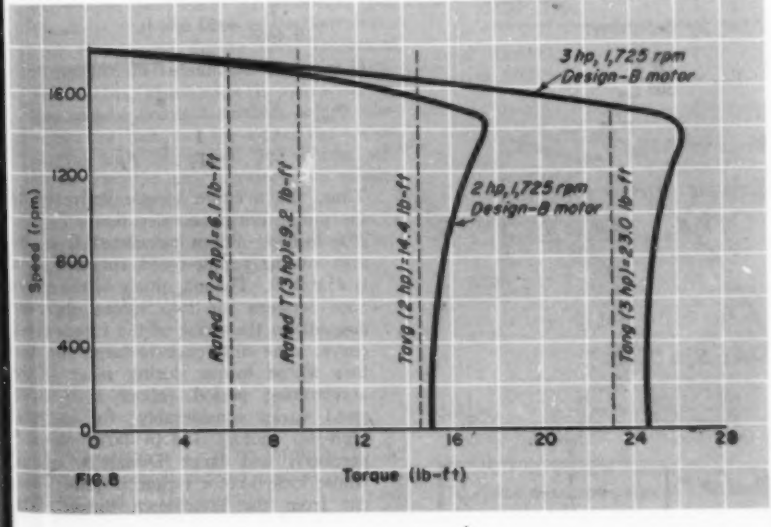


Fig. 6—Duty-cycle of medium size high production lathe originally powered by a 2 hp, polyphase gearmotor. At a production rate of 180 pieces per hr, the motor overheated.

Fig. 7—How torque-speed characteristics of Design-*B* and Design-*D* motors compare during plug reversal. Plugging speeds are shown negative to indicate that torque opposes rotation during the plugging phase.

Fig. 8—Torque-speed characteristics of two hp and three hp four pole motors. These curves are necessary for calculating motor hp from duty cycle.

hp Design-B motor will do the job, the analysis is repeated. In this case, the rms horsepower is calculated to be 3.78 horsepower.

The rms horsepower is increased slightly because the increase in heating caused by plug-reversing the larger inertia exceeds the reduction resulting from increased efficiency of the three hp motor under overload conditions. This result is logical, since the three hp motor is delivering more energy during the fixed period of the duty cycle.

Were a three hp motor used, the ratio of rms horsepower to rated horsepower would be 1.26, an overload of 26 percent. Again, this is sufficient to cause overheating. The obvious solution here is to use a 5 hp motor. A standard 5 hp Design-B motor has a larger frame than either a two or three hp motor. However, if glass insulation is specified, as it was in this case, the 5 hp rating can be obtained in the same frame diameter as the smaller motors. This eliminated the need for making design changes to accommodate a larger frame. Also, there was no necessity for considering the effect of the higher inertia rotor that would be used in the larger frame.

The ultimate criterion of success in motor selection is the actual temperature rise. If the temperature rise of the motor, when it performs the specified duty cycle, is excessive, the machine is undermotored. If the temperature rise is far below rated, particularly if it is below the actual test temperature for rated conditions, the machine is overmotored. Neither condition is perfectly satisfactory from an engineering standpoint.

For certain ratings, field tests on the motor selected for the application will indicate a temperature rise considerably below 40 C for an open motor. In such cases, it will usually be found that the actual temperature rise of the motor even under steady load is below the rated 40 C rise. Motor manufacturers build many ratings with actual temperature rises below rated.

Another point should be mentioned. In effect, the equivalent horsepower method integrates the heating over the entire duty cycle, arriving at an rms value. However, it is possible for the heating during one period to be so excessive as to damage the insulation. This condition would not be evident from the analysis of rms horsepower.

For example, certain machines, such as laundry extractors, have tremendous inertias, with resultant acceleration periods of several minutes duration. Once the machines reach full speed, the load is relatively light. An rms horsepower analysis would indicate that a certain rating is sufficient, whereas it is actually possible for the insulation to become damaged during the long acceleration. These cases are limited to those machines that have wide extremes of load, such as a very long acceleration or a long, heavy overload period followed by a longer, light-load period. The solution is either forced ventilation, an oversize frame, or other mechanical schemes to help dissipate heat during heavy loads.

Another, more precise method of determining horsepower rating from duty cycle is the watts-loss method. This scheme integrates the watts loss for each phase, arriving at the average watts loss over the duty cycle, from which the rating can be selected. The method requires considerable data concerning motor performance and design constants, which is not usually available to the field application engineer.

The accuracy of the equivalent-horsepower method is excellent within its limitations. Its accuracy can be improved by several techniques, all of which add complexity. One, for example, is in the calculation of T_{avg} and t_{acc}. Actually, the average torque

Speed-Increment Method for Obtaining *Tavg* and *tacc*

THE ACCURACY of an equivalent horsepower analysis can be increased by use of this refinement. For some applications, the more positive results obtained may justify the additional time and labor needed for the Speed-Increment Method.

METHOD I—*Where friction is neglected*

Since friction is neglected:

Effective T_{avg} = Effective T_{acc}

Consider t_1 as the time required to accelerate from 0 to 200 rpm, t_2 as the time required to accelerate from 200 to 400 rpm and so on as far as t_9.

ΔS is taken as 200 rpm only because this is common practice. ΔS can be any value, for example, 100 rpm for a 900 or 1,200 rpm motor or 400 rpm for a 3,600 rpm motor.

$$t_1 = \frac{WR^2\,(\Delta S)}{308\,T_1}$$

$$t_2 = \frac{WR^2\,(\Delta S)}{308\,T_2}$$

etc., to t_9. The total accelerating time is:

$$t_{acc} = t_1 + t_2 \ldots . t_9$$

$$= \frac{WR^2\,(\Delta S)}{308\,T_1} + \frac{WR^2\,(\Delta S)}{308\,T_2} + \ldots .$$

$$\frac{WR^2\,(\Delta S)}{308\,T_9}$$

$$= \frac{WR^2\,(\Delta S)}{308}\left[\frac{1}{T_1} + \frac{1}{T_2} + \ldots . \frac{1}{T_9}\right] \quad (a)$$

Also, by definition:

$$T_{avg} = \frac{WR^2}{308\,t_{acc}}$$

or $$t_{acc} = \frac{WR^2}{308\,T_{avg}} \quad (b)$$

Equating (a) and (b):

$$\frac{WR^2\,S}{308\,T_{avg}} = \frac{WR^2\,(\Delta S)}{308}\,\frac{1}{T_1} + \frac{1}{T_2} + \ldots . \frac{1}{T_9}$$

Since $S = 9\ (\Delta S)$:

$$\frac{9}{T_{avg}} = \frac{1}{T_1} + \frac{1}{T_2} + \ldots . \frac{1}{T_9}$$

or $$T_{avg} = \frac{9}{\left[\frac{1}{T_1} + \frac{1}{T_2} + \ldots . \frac{1}{T_9}\right]} \quad (c)$$

Applying this formula to the application

$$T_{avg} = \frac{9}{1/37.0 + \frac{1}{35.5} + \frac{1}{33.5} + \frac{1}{31.5} + \frac{1}{29.0} + \frac{1}{25.5} + \frac{1}{21.5} + \frac{1}{16.} + \frac{1}{7.0}}$$

$$= \frac{9}{0.4465} = 20.6 \text{ lb-ft}$$

If n equals the number of increments:

$$T_{avg} = \frac{n}{\frac{1}{T_1} + \frac{1}{T_2} + \frac{1}{T_3} \ldots \frac{1}{T_n}} \quad (d)$$

Thus, the average torque delivered by the motor during acceleration (T_{avg}) is 20.6 lb-feet. When calculated from the area of the torque-speed curve, T_{avg} is is 25.0 lb ft. The magnitude of the difference between the two values obviously depends on the shape of the torque-speed curve. The difference is large for motors whose torque during most of the accelerating period (about 0 to 1,500 rpm) varies considerably; for example, high-slip motors (particularly above 8 percent), and large Design-B motors, whose locked-rotor torque is quite different from the breakdown torque. The

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obtained from the torque-speed characteristic is only an approximation. To obtain the true average torque during acceleration, a torque-time curve, which is rarely available, is necessary. Or T_{avg} and t_{acc} can be more accurately computed by dividing the torque-speed characteristic into 9 horizontal sections (for an 1,800 rpm motor), one for each 200 rpm increment. The time required for each increment is then calculated from formula (8) in the previous article and these incremental times added to obtain the total. Then, T_{avg} is computed. Because of its length and tediousness, the Speed-Increment Method is employed only for certain applications. See the discussion of this method at the end of this article.

Still another method can be used with a greater degree of accuracy when the data are available. Since heating is a function of in-phase current, the motor rating can be checked by calculating the rms value of in-phase input current. The in-phase input current is determined for each load of the duty cycle and their rms value computed. This value is then compared to the rated in-phase current of the motor tentatively selected. For example, if the rms in-phase current were 7.5 amperes a three hp motor, whose in-phase current at rated load is 8.0 amp, would be satisfactory. This method is rarely applied to motors that start or plug-reverse frequently, since the curves required (watts input or in-phase current input vs time or speed during the acceleration period) are not usually available.

Mechanical Consideration

In selecting motors for severe duty cycle applications, it may be necessary to consider mechanical factors as well as electrical. An extensive discussion of mechanical considerations is beyond the scope of this article since it would involve the complexities of stress-analysis techniques. One such problem might be encountered in plug-reversing of heavy-inertia loads, particularly with larger motors. Depending on the inertia, plug-reversing torque, shaft diameter, frequency of reversal, etc., the question might arise as to whether the rotor is fastened to the shaft securely enough. A stress analysis would answer this and would determine whether special construction were necessary. For most applications, this consideration is neglected.

Other mechanical considerations concern the brushes and commutator for single phase and d-c motors. Where such motors are used to accelerate heavy inertias, for frequent starting and stopping or for plug-reversing, it may be advisable to use an oversize and undercut commutator, and hard brushes to reduce maintenance. In fact, the frequency of operation may be limited by brush life rather than by heating.

These articles have neglected other important but obvious considerations, such as locked-rotor torque and breakdown torque. It is apparent that if the motor selected by an rms-hp analysis does not have sufficient locked-rotor torque to overcome static friction and start the load, a larger rating or a high-torque motor is necessary. Likewise, if breakdown torque is less than peak load torque, larger rating is required.

The purpose of these articles has been twofold: (1) to present a reasonably quick, simple and accurate method for selecting motors for severe service; and (2) to make the reader aware of the type of information required to apply motors for such service. Although the method is a substitute for guesswork, it is not intended to replace good engineering judgment.

In a subsequent article, the author will discuss what part the use of alternates for Design-B motor play in equivalent horsepower analyses. Design-C and -D motors have certain characteristics that must be considered.

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difference is small for motors that give a fairly constant torque during most of the acceleration period; for example, medium-slip motors (up to about 6 percent maximum) and small Design-B motors, locked-rotor torque is practically equal to the breakdown torque. (See torque-speed curves of 1, 2, and 3 hp motor in Figs. 4 and 7.)

Calculation of T_{avg} by the speed-increment method (20.6 lb-ft) gives a lower value than calculation from the area of the torque-speed curve (25.0 lb-ft). The reason is that the area method presumes that the torque during each increment of speed is effective for an equal duration. This assumption is corrected by the speed-increment method, which considers the fact that the larger the torque during each increment, the shorter will be its duration and, therefore, the smaller its effectiveness.

The actual accelerating time (t_{acc}) can be calculated from formula (b) above. Considering WR^2 as 10 lb-ft^2:

$$t_{acc} = \frac{10 \times 1800}{308 \times 20.6}$$

$$= 2.84 \text{ sec}$$

METHOD II—*Where friction is considered*

This method is based on formula (b), by means of which the incremental times are calculated. The method is apparent. From Fig. 9 are obtained the values in this table.

Speed Increment, rpm	Torque lb-ft	Time, sec
0–200	35.0	0.185
200–400	33.5	0.194
400–600	31.0	0.209
600–800	28.7	0.226
800–1000	25.8	0.252
1000–1200	21.5	0.302
1200–1400	16.7	0.388
1400–1600	10.5	0.618
1600–1700	2.5	1.298

Total time (t_{acc}) = 3.672 sec

T_{acc} is therefore:

$$T_{acc} = \frac{WR^2 S}{308\, t_{acc}} = \frac{10 \times 1,700}{308 \times 3.672}$$

$$= 15.0 \text{ lb-ft}$$

The average friction torque during acceleration (approximately 4.0 lb-ft) must be added to T_{acc} to obtain the average torque delivered by the motor.

T_{avg} is, therefore, 19.0 lb-feet.

T_{avg} calculated by Method II is lower than the value obtained from Method I because friction torque increases with speed. Therefore, the durations of the higher values of torque are relatively shorter and less effective.

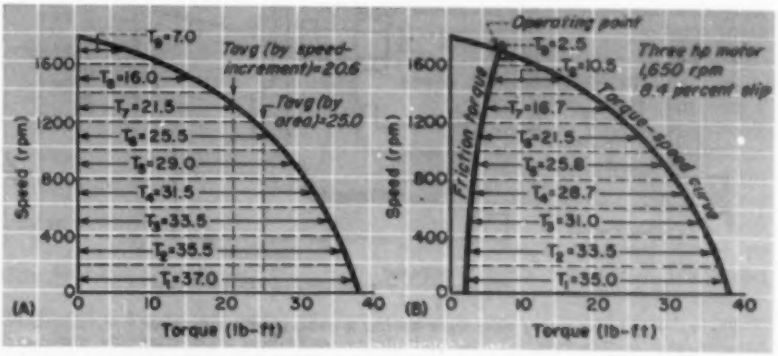

Fig. 9—(A) Typical torque-speed curve used in Method I, where friction is neglected. (B) Same curve prepared for Method II, where friction is taken into account.

Considerably more engineering development has been directed towards methods of reducing the size and weight of aircraft reduction gearing than in gear drives for general industrial or transportation use. It may pay you to take a few tips from the aircraft designer, who solves the problem of reducing gear weight while increasing the power ratings. This can be done by careful consideration of the . . .

. . . Design Factors in Highly L

From information and data furnished by the Wright Aeronautical Corp., Wood-Ridge, New Jersey.

OF THE MANY AUXILIARIES that have been associated with large, aircraft power plants, none has been more important nor received greater attention than the reduction gearing between the propeller and the engine. The recent advent of gas turbine driven propeller aircraft, such as those now being flight tested by Avro in Canada and de Havilland in England, has focused even more attention on this gearing because of the higher reduction ratios and power ratings that are required.

Out of these efforts has come a new design technique for planetary gearing that has resulted in an appreciable decrease in weight, savings in space, and increase in power rating. The major deviation from ordinary planetary gears is the acceptance and provision for deflection in service. As a result, the following factors are representative of the general construction:

1. Use of cantileve mounted planet pinions instead of straddle mounting.
2. Modification of the gear teeth to compensate for operating deflections and thus to minimize pitting and breakage.
3. Use of high contact-ratio gears to increase load-carrying capacity.
4. Establishment of close tolerance limits on: trunnion location, pinion and gear eccentricity, tooth spacing, tooth form, and the backlash variation between pinions.

Types of Planetary Gearing

A planetary gear train is used because it has higher horsepower transmitting capacity and higher reduction ratios per unit weight than most conventional gear trains; for example, the drive shown in Fig. 1 weighs approximately 100 lb. and is capable of transmitting 1750 hp. Planetary gears also have another feature for which they are notable: the ability to provide two

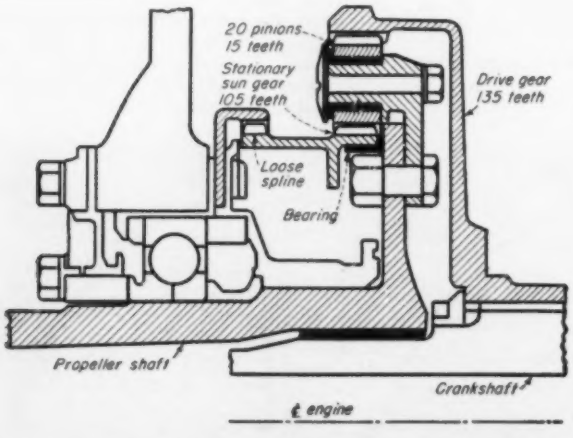

Fig. 1—Simple planetary gear train having a loose spline that takes up lateral motion between gear and pinion teeth, thus equalizing errors in the assembly and insures more uniform tooth loading.

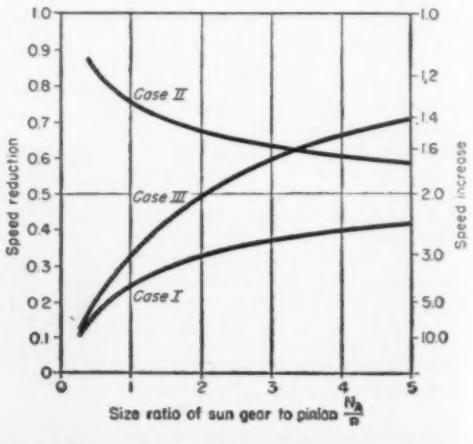

Fig. 2—Speed ratio relationships for simple planetary gear drives in Table I. Except for Case III, the output direction of rotation is the same as input.

Table I—Possible Variations and Speed Ratios of Simple Planetary Gearing

Case No	Function of gearing	Input member	Output member	Reaction member	Direction of output rotation	Speed ratio[2]
I	Speed reduction	A	C	B	Same as input	$\frac{N_A}{N_A + N_B}$
	Speed multiplication	C	A	B		$\frac{N_A + N_B}{N_A}$
II	Speed reduction	B	C	A	Same as input	$\frac{N_B}{N_A + N_B}$
	Speed multiplication	C	B	A		$\frac{N_A + N_B}{N_B}$
III	Speed reduction	A	B	C	Reverse of input	$\frac{N_A}{N_B}$
	Speed multiplication	B	A	C		$\frac{N_B}{N_A}$

[1] Output rotation can be reversed by use of idler gears in contact with each of the pinions.
[2] Letters N indicate number of teeth on various gear members.

Loaded Planetary Gear Trains

or more output speeds at a given input speed without the shifting of gears in or out of mesh. In a simple, single-stage planetary gear drive, this means that two different output speeds can be obtained simply by alternately fixing either the sun gear or the planet carrier. It is this principle that is used in the automatic transmissions in present-day cars.

For instance in Fig. 1, when the sun gear is the stationary member, the gear ratio (reduction) is 0.5625:1 with the power transmitted through the pinion carrier. However, if the pinion carrier were fixed, the ratio (increase) would become 1.286:1 and the sun gear would become the driven member. Such ratio changes are required for aircraft accessories like supercharger drives, where the impeller is driven at one speed during low altitude but an increased impeller speed is necessary to provide sufficient air flow for high altitude flying. In utilizing this method of speed change, the shifting of the reaction member must not be too rapid or high acceleration or deceleration forces would be the result. This speed change is usually controlled by means of a friction clutch, which insures smooth action. Other combinations of speed ratios can be obtained by actually driving the reaction member at a given speed by external means to produce a specific final speed, but this aspect is more complex and beyond the scope of this article.

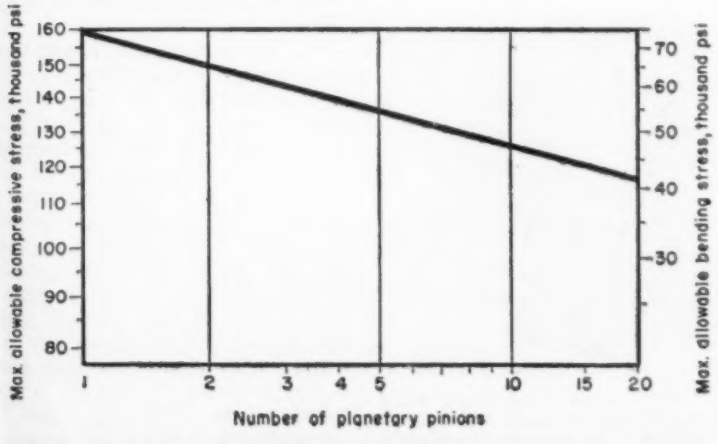

Fig. 3—Number of pinions affects load distribution and must be compensated for. Curve gives the allowable design stress and is conservative as compared to industrial use; derived from empirical data on aircraft drives.

A simple planetary gear train is almost always used when loading is high and when speed ratios are up to 7 or 8 to 1. It consists of three basic members: a sun gear, the planetary pinion carrier, including the pinions, and the enveloping or "ring" gear. Table I shows the possible arrangements of such members, and indicates the manner in which each drive can be used. The arrangement shown in Case I is common for speed reduction, and only occasionally used when an increase in speed is required. For practical reasons this type is limited to a reduction ratio of slightly over 0.4:1, (or output speed is 40 percent of input) and to a speed increasing ratio of around 7:1. A somewhat greater reduction ratio can be obtained from Case III, but the direction is reversed. Case II is used when small speed changes are required. These speed change relationships are shown graphically in Fig. 2.

If a speed ratio greater than 7 or 8 to 1 is required, either two simple planetary gear trains are connected together in series or a compound planetary gear train is used; actually, the speed ratios of most of the high powered, simple planetary drives are in the range of 3 or 4 to 1. The reason for going to a more complex arrangement is that as speed ratios increase, the size of either the pinion or sun gear becomes so small, in proportion to the size of the other members, that

the tooth stresses become prohibitively high and machining becomes more difficult to control. Under these conditions the transmitted loads must be decreased; consequently there is no gain in this direction. However, in the case of the compound planetary train, bearing loads and accumulated errors also introduce a limit on loading; although this type permits almost any speed ratio between input and output. While the compound planetary drive is not as commonly used because of the limitations in the loading, the gear modification discussed in the latter part of this article apply to the compound planetary drive equally well.

Limitations Due To Manufacturing Errors

While the advantage of a planetary gear drive is that a large number of pinions can be used to share the transmitted load, a problem arises with regard to the load distribution among the pinions. As the number of pinions is increased, the ability of the drive to transmit proportionally higher power is decreased because the number of inherent manufacturing errors are increased. Each of the pinions have certain gear tooth errors—however small and within specified limits—that influence the overall performance. These errors consist of pinion trunnion spacing, tooth-to-tooth space, involute profile variations, and variations in parallelism, runout, and backlash among the pinions. To compensate for these errors, the allowable stresses are modified as shown in Fig. 3 and are based on the number of pinions.

This problem of load distribution has long been recognized and a number of methods have been devised to either overcome or minimize the factors that cause overloading of certain members of the drive. One of these methods of load equalization is based on permitting the concentric member to float. This "floating" action allows lateral freedom of one gear so that the net result of the separating forces from each of the pinions determines the operating or running position of the gear. By means of the separating components of the transmitted forces, this centering action prevents excessive overloads on individual pinions.

One such method is illustrated in Fig. 1. In this drive, a loose spline is provided to permit slight lateral motion; however, it is sufficiently restrained to transmit the torque reaction. Another device consists of mounting the gear by means of series of springs, Fig. 4. When these springs are placed around the periphery of the gear, they also serve to "cushion" torsional shock loads. In other instances, the web of the gear is constructed to form a large thin diaphragm that provides the necessary lateral freedom.

In addition to these design devices, the individual gear tooth errors are held to a minimum. The variation in backlash among the many pinions is a factor which considerably influences the proportion of the total load that each one carries. The pinion with the lowest backlash tends to carry appreciably more than its equal share unless the backlash range is held within ± 0.0010 in. The runnout ing of the cantilever trunnions which support the pinions should be held within ± 0.0010 in. The runout for most of these gears is held within 0.0025 in. with a maximum tooth action error of 0.0010 in. To avoid localized stress on the active profile, the involute variation is held within 0.0004 in. The tooth to tooth spacing is maintained within 0.0003 in. The above values apply to reduction gear drives. In general, supercharger drives and gas turbine drives are considered more critical due to the higher velocities (15,000 to 20,000 rpm) and consequently the tolerances are slightly smaller than those specified above.

High Contact-Ratio Gears

To further raise the load carrying ability, it is common practice to increase the contact ratio of the mating gears and pinions. By increasing the ratio of the arc of action to the circular pitch, more teeth are brought into position to share the load, with the result that the maximum load on any one tooth is decreased proportionately.

The contact ratio in ordinary gear teeth is on the order of 1.5. A high contact ratio gear may be defined as one having a contact ratio of over 2:0. The maximum ratio of from 2.2 to about 2.5 is established both by the strength of the teeth and the limitations of manufacture; with too high a ratio, the teeth become pointed, the roots become sharp, and the gearing becomes very difficult to manufacture. To secure a tooth contact ratio of above 2.5, helical gear teeth must be used, but such gearing of course introduces the disadvantages of high axial or thrust loads. To compensate for these added loads the designer is forced to increase the weight of the drive.

Fig. 5 illustrates a pinion with tooth proportions that result in a high contact ratio. Note the long, slender teeth. Although the diametral pitch is 14, these teeth have experimentally transmitted a tangential load of over 700 lb. and at a velocity of 160 ft. per second. The theoretical increase in capacity of this type of tooth form has been verified by extensive testing. With the tooth load always distributed between two or three pairs of teeth, the maximum force which any one tooth assumes (near the pitch line) is approximately 70 percent of the tangential load. This means that for the same compressive stress, the transmitted load can be increased by over 40 percent.

Once the number of teeth have been chosen to provide a particular speed

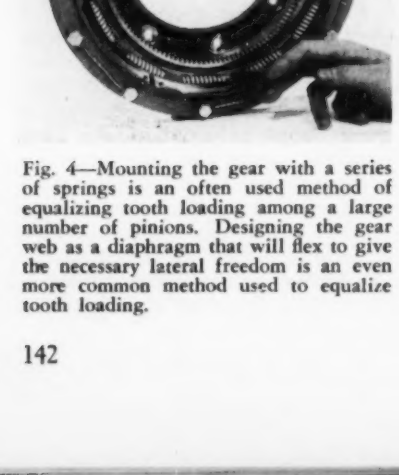

Fig. 4—Mounting the gear with a series of springs is an often used method of equalizing tooth loading among a large number of pinions. Designing the gear web as a diaphragm that will flex to give the necessary lateral freedom is an even more common method used to equalize tooth loading.

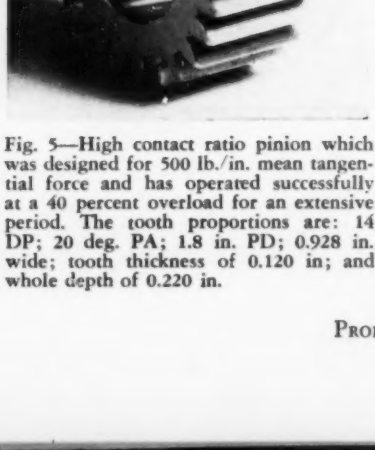

Fig. 5—High contact ratio pinion which was designed for 500 lb./in. mean tangential force and has operated successfully at a 40 percent overload for an extensive period. The tooth proportions are: 14 DP; 20 deg. PA; 1.8 in. PD; 0.928 in. wide; tooth thickness of 0.120 in; and whole depth of 0.220 in.

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ratio, the problem of assembly of the pinions between the two concentric gears arises. To assemble a given number of equally-spaced pinions, P, in a simple planetary drive, the ratio $N_A + N_B/P$ must be an integer. If not, altering the number of teeth in either or both gears A or B should correct the condition. Usually, a change in the number of teeth which is necessary for proper assembly is not great enough to appreciably affect the gear ratio, but this change will slightly alter the pitch diameter and pressure angle. Under these conditions, it is possible for a pinion to have two different pitch diameters; one while meshing with the sun gear and another with the internal gear. In this instance, the sum of the number of teeth in the sun gear and two pinions will not equal the number of teeth in the internal gear. Gear operation under these conditions does not cause adverse operation but is of academic interest. In certain instances when the gear ratio change (to insure proper assembly) becomes excessive, a choice of another pitch is necessary.

A question that often arises is: what is the greatest number of pinions which can be used in a simple planetary drive. The graph Fig. 6 shows this maximum number and is based on the number of teeth in the sun gear and pinions. As the number of pinions is increased, the problem of interference of the pinion teeth becomes evident. Once the number of pinions has been decided upon, the width of the teeth can be calculated. Based on the transmitted load, the pinion width may become relatively large. To avoid this condition, the number of pinions can be further increased by staggering them into two groups. This graph would then be used to determine the number of pinions in each group.

To assemble a compound planetary gear drive, the following conditions must hold: (for equally-spaced pinions)

$$\frac{N_2 n_1 - N_1 n_2}{PF} = \textit{an integer}$$

where: N_1 and N_2 are the number of teeth in the driving and driven gear, n_1 and n_2 are the number of teeth in the mating pinion teeth, P is the number of pinions, and F is the highest common factor between n_1 and n_2.

This equation is valid when the rotation of N_2 is in the same direction as N_1 with the carrier held fixed. The sign in the numerator should be changed (to plus) if the direction of rotation of N_2 is opposite that of N_1 when carrier is held in a fixed position and the driving gear is rotated.

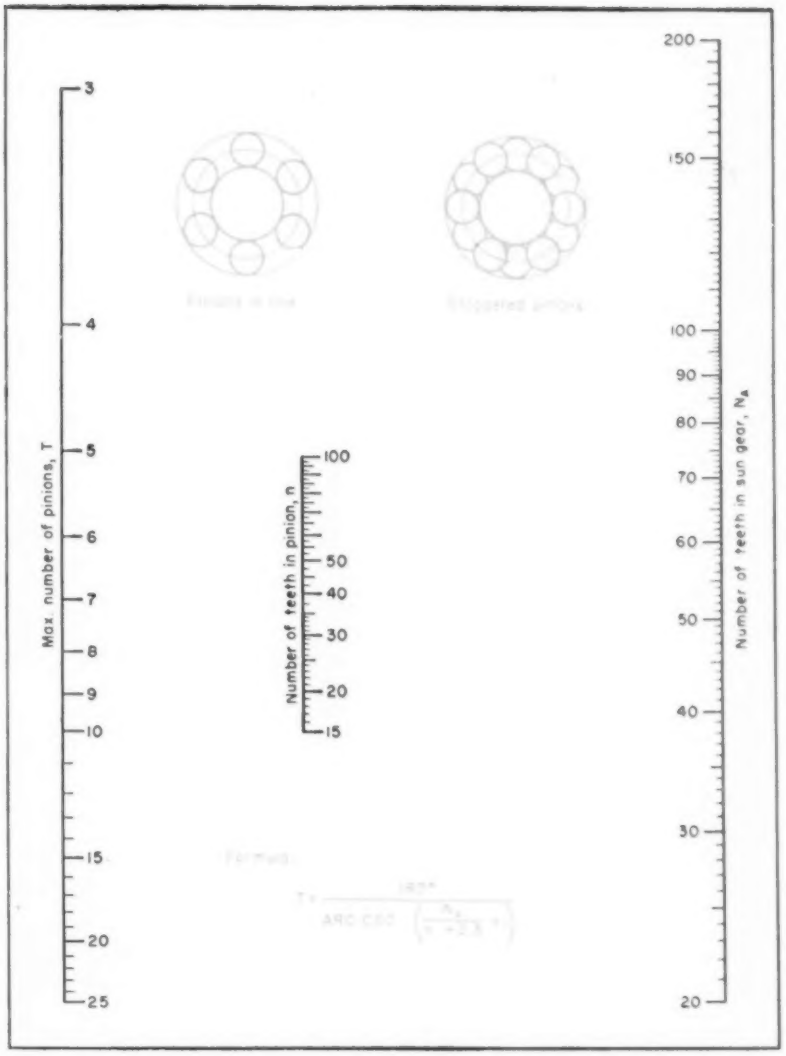

Fig. 6—The maximum number of pinions which can be assembled into a simple planetary gear train without interference can be found by using this nomograph

Applied Forces

The problem of designing gears for reciprocating engines as compared to industrial applications tends to be more difficult. This is largely due to the variations in torque that continually take place as a result of each power stroke. Although elaborate steps are taken to arrange properly the piston cylinders and to dampen the most troublesome disturbances, appreciable torque fluctuations often remain. In this discussion, all reference to loads or forces are considered as the mean value. The problem of determining the actual loads imposed on a gear train is most difficult and can only be performed with elaborate equipment. Rather than introduce a service factor, the allowable values of stress are based on extensive testing where the torque has been found to vary continuously by about ±25 percent. If a load variation is anticipated greater than this value, compensation should be made in the mean value of the transmitted force.

Materials

There are two metals which have been found to be satisfactory for aircraft gearing. The most common is the carburizing steel AMS 6263 (corresponds to SAE 9314) which is case hardened to R_c 60-63, 0.025 to 0.035 in. deep. The other is Nitralloy steel, AMS 6470 (in percent has: 0.40 C.,

0.60 Mn, 1.6 Cr, 0.35 Mo, and 1.15 Al.). It is usually case-hardened to Rockwell 15 N-92 and with a depth of 0.017 to 0.022 in. The Nitralloy material is often preferred for larger gears since a quenching operation is not required to produce the necessary surface hardness. Use of Nitralloy material in shock-load applications is to be avoided, since the very hard case is brittle and prone to crack. In addition, the thickness at the tip of the tooth, of Nitralloy gears, should be greater than twice the depth of case to avoid cracking. With the carburizing steel mentioned, there has been no such difficulty in cases where the tooth was completely carburized.

In using these materials, an allowable surface compressive stress of 160,000 psi has found to be satisfactory. In the case of bending, an endurance limit of 75,000 psi is used. These values of stress are applicable to gears with the tolerances previously mentioned.

Contact Stresses

With the relatively large ratios that often exist between the pinion and the mating external gear, the compressive surface stresses are a critical factor. These stresses are determined by the Hertz equation which is based on cylinders in contact. These contact stresses rapidly increase as the tooth action approaches the base circle since one of the radii of curvature then approaches zero.

The compressive stresses within the length of action of standard, equal addendum gears are illustrated in Fig. 7(A). Note the rapid rise in stresses near the pinion base circle. Increasing the pinion addendum and reducing the addendum of the gear alters the tooth action to a region where the stresses are much lower. The problem of compressive stress is much less critical when the size ratio between two mating gears approach unity, as shown in Fig. 7(B), but under these conditions, the bending stresses become the limiting factor in proportioning the size of gear teeth.

To emphasize the variation in compressive stresses, Figs. 7 (A) and (B) were based on the condition that the full load is carried by one pair of teeth throughout the line of action. This condition would exist only when the contact ratio is equal to 1.0, which is to be avoided in good gear design. Consequently, the stress must be modified to take into account the magnitude of the actual force being transmitted between mating teeth during the period of contact.

The manner in which the load on a particular tooth would vary throughout the length of action if the teeth were absolutely rigid is also shown in Fig. 8. It can be seen that the entire load would be assumed by one pair of teeth only in the region from b to c. Actually, the teeth are by no means rigid and if the elasticity or spring constant of a mating pair of teeth were to be determined for various positions of contact, it would resemble the dotted curve shown. In other words, when contact occurs near the pitch line, a greater torque would be required to deflect one gear with the other gear fixed, as compared to any other position of contact. It is to be noted that this estimated elastic curve approaches zero at each end; it is recognized that a relatively small amount of the total load will actually be transmitted in this region. This estimated curve is satisfactory for all practical purposes since the stresses are not appreciable at the ends of action unless there is on excessive tooth spacing error.

In general, computing these stresses at the pitch point gives a good indication of the possibility of pitting or wear. In special instances, the stresses should be investigated throughout the length of action when they are apt to be appreciably higher than at the pitch line. The compressive stress in the usual wear formula is computed at the pitch line:

For external steel gears

$$S_c = 4580\sqrt{\frac{F}{W \sin 2\alpha}\left(\frac{P_1 + P_2}{P_1 P_2}\right)}$$

where: S_c is the compressive stress, psi; F the tangential force, lbs; W is the width of tooth, in.; P_1 and P_2 are the pitch diameters of gear and pinion, in; α is the pressure angle, deg.
Note: for internal gears, change the positive sign in brackets to negative.

For convenience, a nomograph has been prepared to determine the compressive stress of gear teeth. This graph is based on the number of teeth in the gears, and takes into account the pressure angle of the gear teeth. See pages 159 and 161.

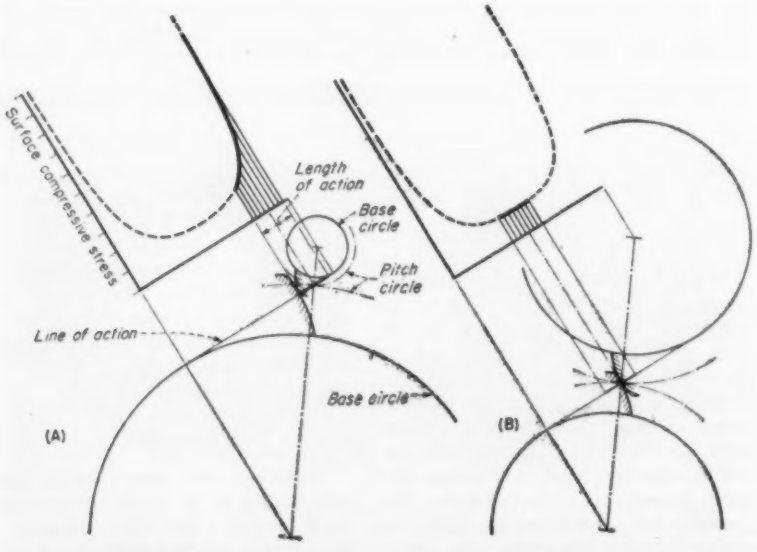

Fig. 7—Comparative surface compressive stress of mating gear teeth: (A) teeth having equal addendums; (B) lower stress of equal-size gears. These curves are based on the assumption that one tooth is carrying the full load.

Bending Stresses

With the relatively large forces and the high velocities which are present on many planetary gear teeth, the possibilities of tooth failures by breakage are high. Under these conditions, it is extremely important to avoid any increases in bending stresses at the roots of the teeth. Two factors which significantly influence this stress are: (1) Impact loading that is produced by either a pulsating load or high peripheral speeds on gears having excessive tooth errors; and (2) stress concentrations at the tooth root due to a relatively small root radius.

In computing the bending stresses, the following formula has been found to be satisfactory:

$$S = \frac{1.5\,FK}{XWCR}$$

where F is the pitch-line tangential force applied at the tip of the tooth,

X is the strength factor, K is the stress concentration factor, CR is the contact ratio, W is the tooth width.

This equation is basically the Lewis formula with certain modifications. Since special tooth proportions exist on most aircraft drives, the Lewis Y factor from the standard tables cannot be used; instead, the X distance or tooth strength factor is obtained from an enlarged tooth layout. (For standard-proportioned gears, this factor is used to determine the Y factor, $= \frac{2}{3}X$). Stress concentration factor K is based on photoelastic tests and is defined in Fig. 9. The main difference between this modified equation and the usual Lewis Formula, is that the velocity co-efficient which greatly decreases the allowable tooth load is neglected. Elimination of this factor is possible since gear tooth errors are controlled within close limits; hence, the impact or increment loading is decreased and higher design loads are permitted.

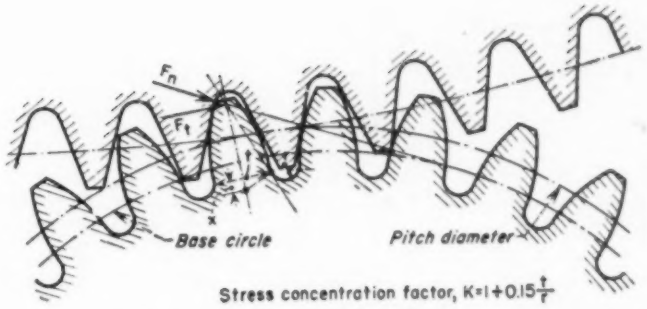

Fig. 9—The tooth strength factor, X, can be graphically determined by an enlarged tooth layout as illustrated. The importance of a sharp fillet radius is accounted for in the stress concentration formula given above.

Sliding Velocity

This factor in gear design does not present any problem in most aviation gear drives. This sliding velocity tends to generate friction especially at the extreme portion of the length of action near the tips or flanks. Since most drives are of relatively small pitch, the magnitude of sliding does not reach high values even in high-speed supercharger drives. Sliding velocities of 60 fps have successfully been operated. The sliding velocity can be calculated as follows:

For external gears

$$V_s = 2Va\left(\frac{P_1 + P_2}{P_1 P_2}\right)$$

where: V_s is the sliding velocity, fps; V is the pitch line velocity, fps; a is the distance along line of action from pitch point to extreme point of contact, in; P_1 and P_2 are the pitch diameters of gear and pinion, inches. For internal gears, the sign in bracket is negative.

In each new gear drive a considerable amount of development work is required before the drive is considered as satisfactory. Although calculations indicate the direction and approximate magnitude of operating deflections which tend to cause uneven gear tooth loading, actual tests are relied upon to determine finally the necessary tooth modifications that are required for optimum performance. Since most of the planetary pinion supports are of the cantilever type, the teeth must be altered to insure full width or contact under full power. This type of modification consists of machining the teeth out-of-parallel and is usually performed on the concentric ring and "sun" gears while the teeth of the pinions are closely held parallel to the axis of rotation. This modification is performed only on the concentric gears since only one side of the teeth on each of these gears need be controlled as compared to having to control both tooth surfaces on the many pinions which are used. In reduction drives, parallelism of the pinion teeth are held as close as 0.0003 in. per in.; in the case of the internal and sun gears, the mean value of out-of-parallel has been as large as 0.0020 in. per inch.

Although the basic involute gear tooth form is used, the effect of large forces causes tooth deflections that alters the original profile. Thus involute modifications are required to insure conjugate tooth action. These deflections, unless corrected, result in scuffing, gouging or pitting. In many instances, the flanks of pinions are appreciably relieved to decrease the load and consequently lower the compressive stress, which tend to become high in the region near the point of tangency of the line of action and the base circle. As a general rule, in high capacity gears the involute form of the pinion flank is relieved appreciably while the mating gears are held to the basic involute form.

This type of tooth modification has been treated in several different ways. In the Feb. 1947 and Sept. 1949 issues of PRODUCT ENGINEERING, methods of tooth modification were described.

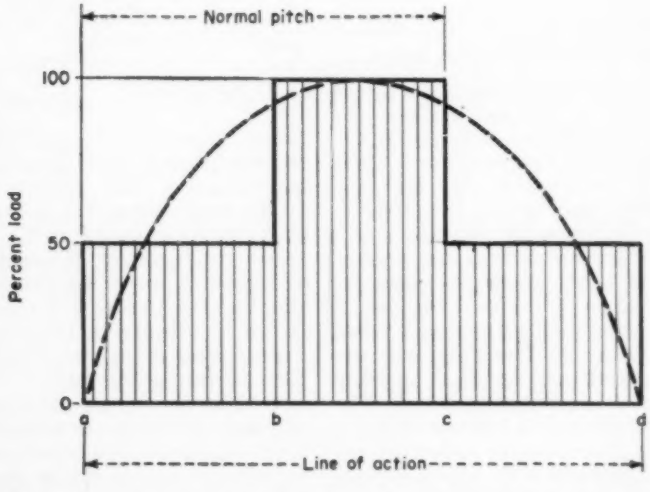

Fig. 8—Force on a gear tooth is not constant but fluctuates as teeth go in and out of mesh. This variation is illustrated for gears with 1.5 contact ratio; for inelastic teeth (full lines) as well as flexible teeth (dash line).

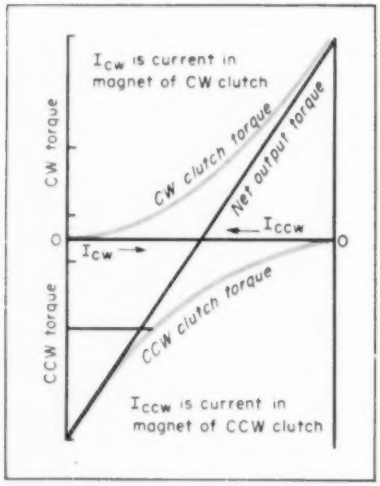

Fig. 1—Characteristic curves of a magnetically actuated dry friction clutch made of two clutches opposing one another.

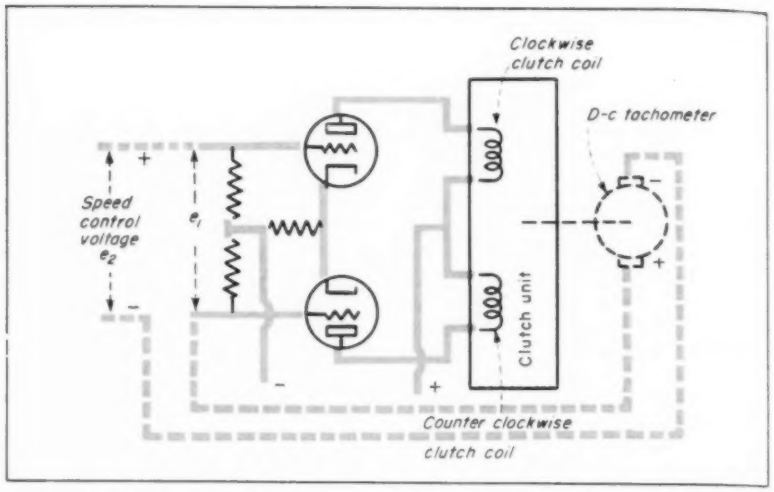

Fig. 2—Triode vacuum tubes are a convenient way of controlling magnet current. Solid lines show the basic circuit for controlling the clutch coils by signal e_1; dotted lines indicate the necessary circuit additions to obtain a velocity servomechanism.

The Dry Friction Clutch As A Servomechanism Drive

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FRICTION CLUTCHES AND BRAKES have long been used for starting and stopping rotating machines, but it is only recently that friction torques have been controlled so that clutches and brakes can be made to deliver continuous smooth power at an intermediate speed. Various rolling friction drives have been used as torque converters, but these devices are generally difficult to control and they have not proved advantageous for servomechanism drives.

This discussion will consider only magnetically actuated dry sliding friction clutches, similar to those commercially available for controlling machine tools. A pair of these clutches can be arranged so as to drive a load in either direction, depending on which clutch is excited. The resulting device makes a high performance servomechanism. It can be made to give accelerations of 500,000 radians per sec^2, and has a frequency response characteristic which is still unity at twenty cycles. It is smaller and lighter than any other type of servomechanism in the power range from twenty to five hundred watts and is simply constructed.

Clutch Unit Characteristics Needed for a Servomechanism Drive

A simple friction clutch is a torque device having no speed stability. The output shaft either runs at the full speed of the driving motor, or it stands still. No fixed value of force will cause the output shaft to turn at half speed.

An electromagnet provides a convenient way to obtain a continuously variable, rapidly controllable force for clutch actuation. This combination of a clutch and an electromagnet is called a clutch section. Two such clutch sections which are operated opposing each other is called a clutch unit. The torque output of a cluch unit is the difference between the currents in the two magnet coils if the sum of the two currents is maintained constant. In Fig. 1 this is shown graphically.

Control of the magnet current is easily obtained with two triode vacuum tubes. The circuit of Fig. 2 (solid lines) is typical. The torque is now a linear function of the control signal e_1, changing direction as e_1 changes sign, and is a good torque source for a servomechanism drive. For servomechanism analysis, it can be represented by the simple transfer function:

$$T(p) = \frac{Ke_1(p)}{(1 + tp)}$$

Letter t represents the time delay in seconds between input signal e_1 and output torque. This delay is caused by the inductance of the magnets.

It should be noted that this torque source is completely independent of the speed at which the clutches are slipping. For a position servomechanism, it is customary to have the driving members of the two clutch sections rotating at the maximum speed desired, but in opposite directions. If, however, a velocity servo-mechanism were desired, one clutch section driving member could be at zero speed and the other at maximum speed. In either case, the same torque transfer function would be used in the performance calculations.

The lack of any speed sense in the

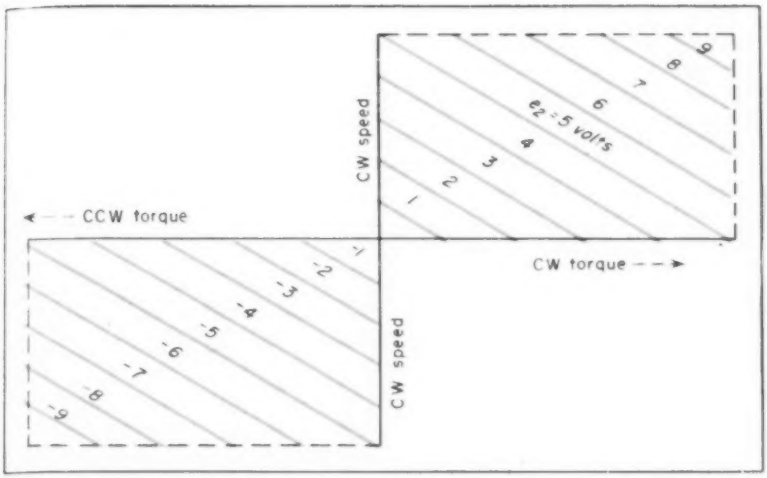

Fig. 3—The speed-torque characteristics of a friction clutch velocity servomechanism are similar in form to those of a d-c shunt motor where e_2 corresponds to the armature voltage and the tachometer voltage corresponds to the back emf of the motor.

torque source means that a position servomechanism using the source would have no damping whatever. Of the several methods for providing damping, tachometer feedback is the most practical. In Fig. 2 a d-c tachometer generator is coupled directly to the output shaft of the clutch and its output is added to the control signal e_2 in such a direction that it cancels a portion of e_2 when the output shaft turns. The combination of clutch unit, current control, and tachometer is now a velocity servomechanism. The curve of Fig. 3 shows the speed torque characteristic for this servomechanism. This should be recognized as the speed-torque characteristic of a d-c shunt motor, where e_2 corresponds to the armature voltage and the tachometer voltage corresponds to the back emf of the motor. Considering the output torque zero, the clutch applies torque to accelerate the shaft until it reaches the speed where the tachometer voltage just equals the input voltage, at which time e_1 is zero and no further torque is applied.

A stable position servomechanism can be made from the velocity servomechanism by making e_2 proportional to the position error. The block diagram of Fig. 4 shows such a servomechanism. Each block corresponds to a transfer function: K_1 to the torque source, K_2 to the shaft inertia, K_3 to the tachometer, and K_4 to the position error amplifier. By using either the linear or logarithmic procedure, the frequency response of such a clutch servomechanism can be calculated, Fig. 5. Test frequency responses have agreed fairly well with the calculation for a servomechanism with tachometer feedback. The calculated performance of the clutch stabilized with error lead networks could not be attained in practice because the lead networks increased the noise-to-signal ratio so that the amplifiers saturated.

It might be thought that the motion of the output shaft is jittery, since two clutches are trying to turn one shaft in opposite directions and since friction is never very consistent. Actually this jitter would exist if the clutch unit were not properly designed.

Tests have shown that variations in output torque are of three kinds: (1) Variations taking place over a period of minutes or hours caused by changes in the average coefficient of friction between the sliding surfaces, (2) cyclic variations having a frequency corresponding to the slipping speed; and, (3) random frequency pulses of torque of very short duration, which are inherent in dry friction.

The first of these variations is so slow that it is easily corrected for by the regulating properties of the closed servomechanism loop. In any practical clutch servomechanism the second variation is always at too high a frequency to be smoothed out by regulation, so it becomes necessary to make this rotation frequency so high that the clutch shaft, in spite of its low inertia, acts as a flywheel to average out these torque changes which may amount to as much as 20 percent of the maximum output torque. Clutch speeds from 2,500 to 3,600 rpm, corresponding to frequencies of 50 to 60 cycles per sec, have been found to be best. The third source of torque variation is made up of even higher frequencies and is damped out in the same way. Thus on a practical clutch position servomechanism, when the error

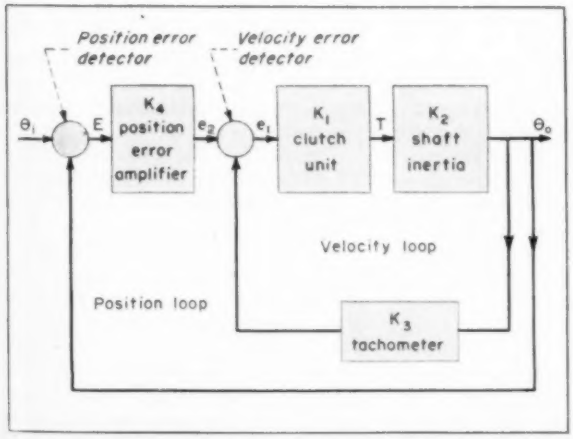

Fig. 4—Block diagram of a position servomechanism with tachometer feedback used to obtain stabilization.

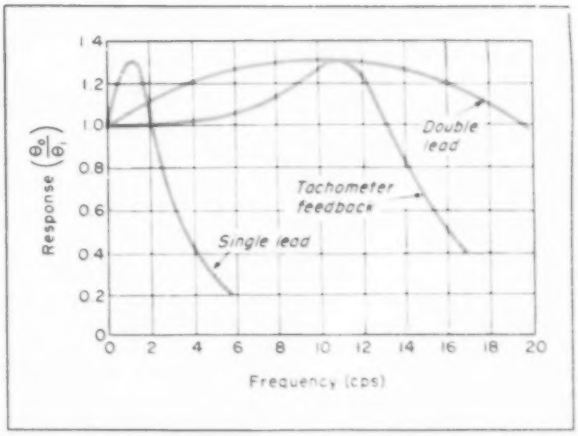

Fig. 5—Curves representing the calculated frequency responses of a ½ hp clutch for several methods of stabilization.

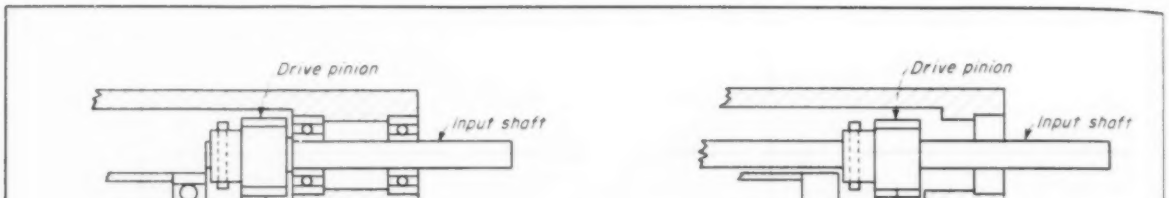

Fig. 6—Clutch sections with a single friction surface and utilizing: (*A*) a rotating magnet; (*B*) a stationary magnet.

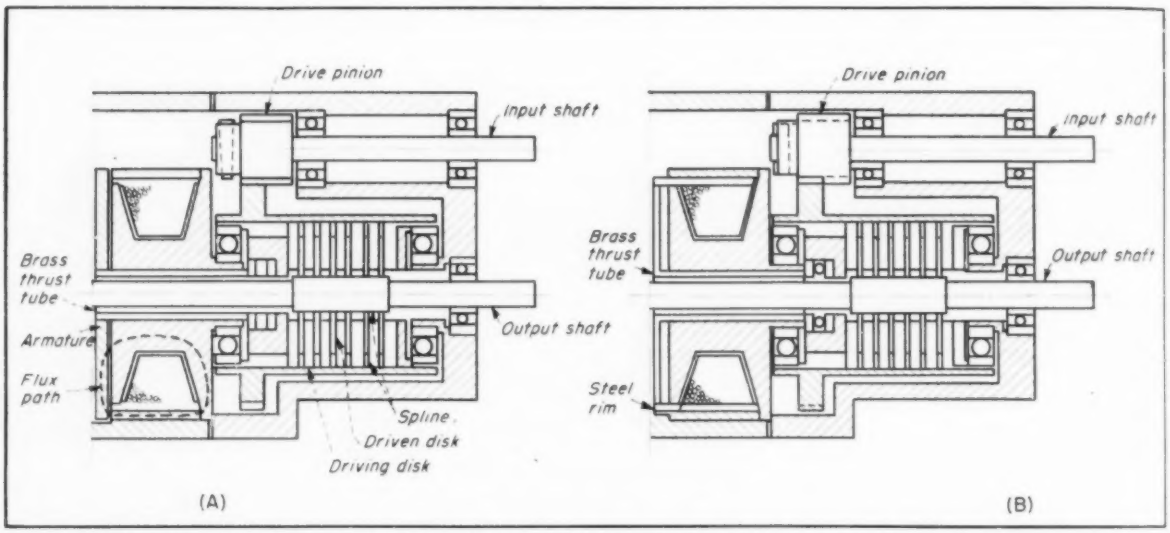

Fig. 7—Clutch sections with multiple friction disks and utilizing: (*A*) a flat-faced magnet; (*B*) a solenoid-type magnet.

signal is zero, the output shaft has very little motion.

Types of Clutch Sections

Clutch sections are of two general types: Those having a single friction surface as in Fig. 6 (*A*) and 6 (*B*) and those having multiple friction discs such as Fig. 7 (*A*) and 7 (*B*) show. Each of these divisions can be further broken down according to the type of magnet used, and the method of applying the magnet force to friction surfaces which are rotating. While a large number of combinations are possible, only four have special advantages.

The particular advantage of the single surface clutches is that they maintain their torque as the friction surface wears. This advantage occurs because the gap across which the magnet force is developed is between the friction surfaces. This means that the driven member must be of magnetic material and must be thick enough to carry the flux of the magnet. Thus the choice of friction materials is restricted, and the acceleration which can be attained by the driven disc is limited, since its inertia is a function of the output torque.

Even though these two disadvantages exist, servomechanisms have been built using this type of construction. The magnet may either rotate and have its current supplied through slip rings or be stationary with the flux crossing an airgap. The need for brushes can be considered a disadvantage of the first type, while the stationary magnet type requires additional excitation power for the airgap.

The multiple disc type clutch in Fig. 7 (*A*) permits an unlimited choice of friction materials since the friction surfaces are separate from the magnetic circuit. However, in order to obtain this separation, the magnet must develop its force across an airgap of some length. The construction of Fig. 7 (*A*) utilizes a flat-faced magnet whose force is transmitted to the rotating clutch discs through a thrust bearing. It is easily seen that this force increases as the disc stack wears until the magnet airgap goes to zero and no

force is applied to the disc stack. In cases where a life of less than 20 hr can be tolerated, this method of construction will give a servo drive with the highest acceleration and power gain for a given size and weight.

If a longer life is desired, the magnet construction of Fig. 7 (*B*) is better since the force can be made independent of wear for as much as one-eighth inch of armature movement. For a given size and current, this type magnet has approximately one-fifth the force of the flat-faced magnet.

Another source of force for an extremely short time delay is the dynamic speaker motor. This requires a very large and heavy field structure for useful magnitudes of force. It has a much lower inductance and produces a force which is a linear function of current.

Mechanical Considerations in Clutch Unit Construction

The construction of a reliable clutch unit is largely a matter of good mechanical design. Since a clutch unit must have two barrels which rotate in opposite directions, at least three spur gear meshes, two shafts and 8 bearings are required. In a typical clutch the input will be at 10,000 rpm. A pinion at this speed drives the barrels at 2,500 to 3,600 rpm. This high speed gearing requires good lubrication such as is obtained by completely enclosed gear cases. Precautions must be taken to see that none of the grease from the gears or bearings reaches the clutch disc surfaces. All gearing should be of hardened steel and the alignment must be carefully maintained by center fits and doweling.

Clutches have been built with oilless bronze bearings or with ball bearings throughout. Experience has indicated that the ball bearings are more reliable at the high speeds. In addition, bronze bearings when used for the barrels waste a considerable amount of power. Bronze bearings are satisfactory for the output shaft because it runs at a very low average speed in a position servomechanism. It has been found that the ball thrust bearings when run dry will outlast the friction materials. Dry operation is almost a necessity because of the close proximity of the bearing to the friction surfaces which reach temperatures of 500 F approximately. A heat insulator, such as is shown in Fig. 7(*A*), reduces the bearing temperature considerably, but it is still too high for lubrication with petroleum greases.

The friction materials are best evaluated by tests which duplicate operating conditions as nearly as possible. Published data on coefficient of friction and wear have been found to be of little use in determining output torque and life. The high temperature throughout the entire clutch disc stack limits the discs to inorganic materials. Stainless steel against carbon or bronze have a long enough life for most applications. There is a great deal of work yet to be done to determine the best materials for this type of service.

Comparison of a Clutch Servomechanism with Other Types

A clutch servomechanism capable of 100 watts output can be made smaller and lighter than any other type. The main reason for this is its simplicity.

In Table I are tabulated typical component parts and their weight and space requirements for different types of servomechanisms of the 100 watt size operating from a 24 volt d-c supply. The table gives residual losses of the systems when the drive is not producing torque. The clutch servo compares favorably in this respect.

The clutch servomechanism is reliable. Here, reliability is defined to mean that the servomechanism must have exactly the same performance every time it is turned on without any adjustments during the life of the device. This definition admits the possibility of building a short life device whose chances for failure during its life are just as small as the chances of failure of a long life device operating for the same length of time. In the case of the clutch, this is done by developing the clutch unit to the point where failure occurs only because of wear of the friction surfaces. It is then only necessary to determine the minimum life of the friction material to know how long the device is 100 percent reliable.

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3. **Electronics Instruments (Book), Greenwood, Holdam, and McRae, MIT Radiation Laboratory Series, McGraw-Hill Publishing Company, Pages 393 to 404.**
4. **"Electronic Auto-Pilot Circuits," W. H. Gilli and H. T. Sparrow, Electronics, October 1944.**

Table I—Weight of Typical Servomechanism for 24-Volt D-c Operation

Servomechanism Type	Component	Weight, lb.	Volume, cu in.	Loss,* watts
Two Phase Motor Vacuum Tube Control	Inverter	18	225	
	Motor	11	150	
	Preamplifier	2.5	80	
	Gearbox	3	45	
	Tubes	1.5	40	
	Output Trans.	9	90	
	D-C Supply	15	250	
	Total	60	910	267
Two Phase Motor Thyratron Control	Inverter	18	225	
	Motor	11	150	
	Preamplifier	2.5	80	
	Gearbox	3	45	
	Thyratrons	1.5	36	
	Output Trans.	9	90	
	Total	45	626	272
Two Phase Motor Saturable Reactor Control	Inverter	18	225	
	Motor	11	150	
	Reactors	2.5	25	
	Rectifiers	.5	15	
	Preamplifier	2.5	80	
	Gearbox	3	45	
	Total	37.5	540	240
D-C Motor Thyratron Control	Inverter	18	225	
	Motor	4	57	
	Preamplifier	2.5	80	
	Gearbox	3	45	
	Thyratrons	3	72	
	Total	30.5	459	234
D-C Motor Ward Leonard Control	Motor-Generator	10	120	
	Motor	4	57	
	Preamplifier	2.5	80	
	Gearbox	3	45	
	Total	19.5	282	95
Hydraulic Motor	D-C Pump Motor	4	45	
	Pump	4	40	
	Solenoid Valve	2	30	
	Motor	4	40	
	Sump and Oil	5	350	
	Preamplifier	2.5	80	
	Gearbox	3	45	
	Total	24.5	630	95
Clutch	D-C Motor	4	45	
	Clutch Unit	3	40	
	Preamplifier	2.5	80	
	Gearbox	3	45	
	Total	12.5	210	110

* For zero torque output.

ENGINEERING ABSTRACTS

FOREIGN AND DOMESTIC ARTICLES

Effects of Boron On Cast Iron

From "Effect of Boron on the Structure and Properties of Cast Iron", Technical Report No. 1362, National Bureau of Standards, U. S. Department of Commerce, Washington 25, D. C.

WAR-INDUCED SCARCITIES of other elements called attention to the possible use of boron to increase the hardness and wear resistance of cast iron. However, specific data regarding the effect of boron on the structure and properties of cast irons have been somewhat limited.

In the Bureau's investigation, plain gray cast irons of nine different stocks were studied. The carbon content in these irons varied from 3.3 to 4.1 percent, the silicon content from 1.4 to 2.7 percent. The irons were melted in high-frequency induction furnaces, and boron in the form of ferro-boron was added to the molten metal in amounts ranging from less than 0.001 percent up to 0.48 percent.

When examined under a microscope, the structure of irons with a relatively large boron content was found to consist essentially of a matrix of cementite containing islands of pearlite as well as other structureless islands. The pearlite areas were usually surrounded by a structureless band.

Transverse strength was determined by subjecting bars 1.2 in. in diameter to a breaking test. A disk $\frac{3}{4}$ in. thick was then cut from each broken transverse test bar adjacent to the fracture and used for Brinell hardness determinations and for metallographic examination. Fractures could be classified as gray, white-network or mottled. For bars with gray fractures, an increase in boron content was accompanied by a small increase in the transverse strength, Brinell hardness, and relative modulus of elasticity of the sample. However, the appearance of a white network or white specks on the fractured surfaces of bars containing larger amounts of boron usually was accompanied by a drop in the transverse strength and deflection, and an increase in the Brinell hardness and relative modulus of elasticity. Brinell numbers varied from 140 to 220 for bars with gray fractures, from 220 to 285 for bars having a white network, and from 295 to 427 for those with mottled and almost white fractures.

Another very important property of cast iron is its ability to chill with the formation of white iron. For all castings made under similar melting and pouring conditions, there was a pronounced increase in depth of chill with an increase in the boron content.

Chilled areas usually consisted of two zones: a completely white zone, known as "clear chill," and a partly chilled or mottled zone.

In general, the results of the investigation show that boron increases the depth of chill of cast iron as well as its hardness. Boron was also found to decrease the size and quantity of graphite particles, to increase the free carbide content, and to produce an undercooling effect which becomes more pronounced with greater percentages of boron.

Synthetic Piezo-Electric Materials to Replace Quartz

Foreign abstract from "Piezo-Electric Materials" by J. C. B. Missel, Philips Technical Review, Vol. II, No. 5, pp. 145-150, November, 1949.

IN PIEZO-ELECTRIC MATERIALS, mechanical stress gives electrical effects. Applications have been extended from sensitive instruments to relatively powerful industrial devices. New materials have been developed to replace quartz as a piezo-electric material.

Under a mechanical load, a quartz crystal has electrical charges of opposite polarity on its opposed faces. The converse is also true; crystals show mechanical deformation in an electric field. Both effects are linear. The resultant charge is directly proportional to deformation, and mechanical deformation is proportional to the applied electric field strength. This is entirely different from electrostriction, in which any substance, crystalline or amorphous, is deformed in an electric field with a displacement which varies as the square of the applied field strength.

The strength of the piezo-electric effect depends mainly upon the material and the direction of the mechanical load. The effect is very strong in the case of seignette salt (Rochelle salt, potassium sodium tartrate). In a certain direction of the crystal, a charge can be obtained amounting to about 8×10^{-9} coulomb per newton of force applied (2.3×10^{-4} esu/dyne). With quartz the charge is 2×10^{-12} coulomb per newton.

One use of the piezo-electric effect is for indicating mechanical vibrations by electrical means and, conversely, for generating mechanical vibrations by means of alternating voltages. The waves emitted by supersonic radiators are used for signalling under water, for preparing very finely distributed emulsions and suspensions, and for tracing defects in large pieces of material. Piezo-electric crystals are also used for stabilizing the frequency of radio transmitters.

With one single exception, piezo-electric effects may be expected in all species of crystals not exhibiting a centre of symmetry in their crystal lattice. If a crystal has a centre of symmetry, each positive ion is balanced by another positive ion, exactly opposite to it through the center of symmetry, at an equal distance from the center. The same is true of negative ions. The center of gravity of the positive ions and the center of gravity of the negative ions are both at the center of symmetry of the crystal.

A piezo-electric crystal, on the other hand, has no center of symmetry. In the natural state, the center of gravity of the positive ions and the center of gravity of the negative ions are at the same point. When the crystal deforms, however, the two centers of gravity are displaced and move in different directions. They are no longer at the same point and the crystal becomes an electric dipole. The converse takes place when an outside field is imposed on the crystal.

About 1,000 of the 10,000 crystals so far identified have no center of symmetry and so exhibit the piezo-electric effect. For practical applications, large crystals are essential, and only a few of the possible 1,000 can be considered. Zince blende is difficult to work and readily splits; tourmaline crystals are hard to obtain with a sufficiently homogeneous structure. Quartz is the only practical natural piezo-electric crystal.

Attempts have been made to produce synthetic piezo-electric materials in sufficiently large crystals. Ammonium dihydrogen phosphate, dipotassium tartrate and ethylene-diamine tar-

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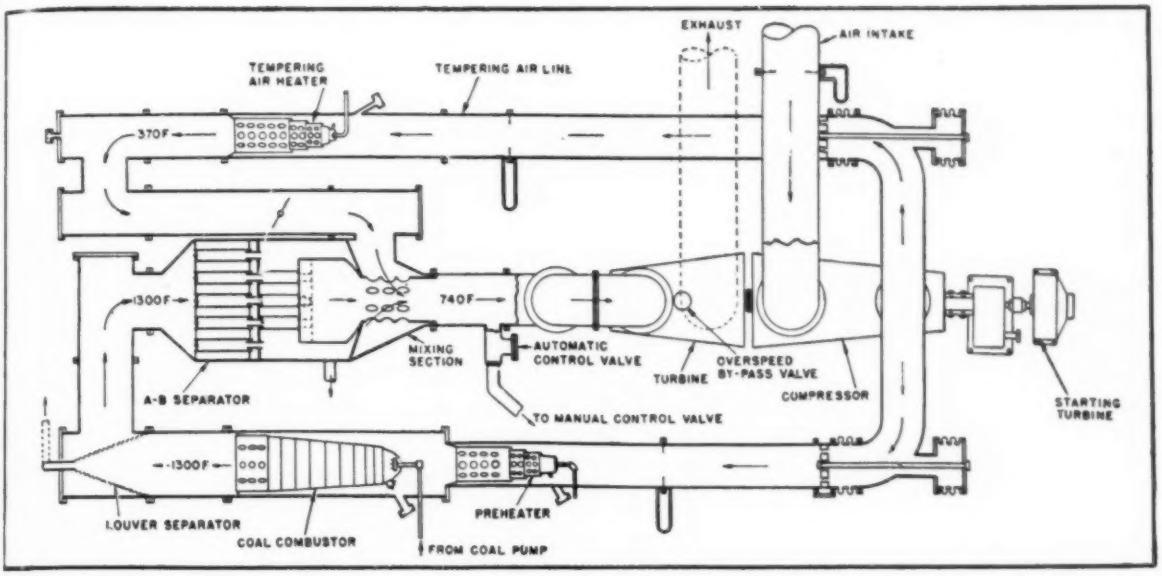

COAL IS BURNED on one side of locomotive-size test plant, oil on the other. Real locomotive will burn coal on both sides, operating turbine at full 1300 F temperature. Two separators extract fly ash before combustion gases enter the turbine.

Coal-Fired Gas Turbine

Abstracted from "The Coal-Fired Gas Turbine" Coal Age, May, 1950.

IN A TEST STAND OPERATION in December 1949, The Locomotive Development Committee, Bituminous Coal Research, Inc., ran a locomotive-size gas turbine, using pulverized coal as fuel. Coal handling, combustion and flyash removal problems have been solved after five years of research and development.

Coal is pulverized and stored in a bin from which it is fed to a pressurizing rotary pump. The air compression system supplies 220,000 lbs. per hour of which 93,000 goes through the coal-burning component and the remainder to an oil burner. A preheater on the coal side provides a boost that later will come from the locomotive regenerator.

Burning takes place in a cylindrical, stainless steel combustor, where the temperature of combustion products and excess air jumps to 1300 F. Combustion efficiency exceeds 95 percent at 1,800 lb. of coal per hour. At this firing rate, heat release is 750,000 Btu per hour per cu ft, or 6,000,000 Btu per hr per sq ft of cross sectional area. There is relatively little slagging and nearly all the ash appears as a fine dry powder suspended in the combustion gases.

The oil burners are similar to film-cooled combustors used on jet aircraft, and the discharged air is used to temper the far hotter air from the coal side.

Fly ash is removed from the coal-side gases in two steps. A lower separator concentrates the large particles in a small air stream in the first stage. This stream is then blown down to atmospheric pressure, quenching incandescent coal particles.

The second stage removes the remaining 50 percent of original solids in two banks of American Blower tube separators. Ash concentrates into 10 percent of the air stream in the first bank, and is packed into 1 percent of the original air in the second. This dust-laden air is exhausted.

The turbine in this test could not tolerate temperatures over 1000 F, and the coal combustion gases were cooled after the fly-ash separator, by mixture with the oil section exhaust. Resulting temperature is 740 F. Actual locomotive turbines will use gas at full coal combustion heat, 1300 F.

The turbine is started by an auxiliary steam turbine and speeded up by the oil side before the coal combustor is started. Rated speed is 5,180 rpm. Speed holds fairly constant, though temperatures at the combustor outlet vary as much as 25 percent from mean values.

No loads have been put on the turbine as yet, but application to a 1000 KW alternator is contemplated. After 250 hours of operation no damage or efficiency drop from fly ash was noted, although some localized erosion of stationary blades took place.

The experimental locomotive is on the drawing boards and Allis-Chalmers has completed and tested a 4200 hp, 1300 F turbine. An 1275 F Elliot turbine is being designed. If further tests are satisfactory, the Allis-Chalmers turbine will be on the locomotive in early 1951.

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trate have been developed. They are shaped easily, have small internal damping, and are not too readily dissociated. Piezo-electric constants 6 to 8 times those of quartz have been reached. The ratios of capacities C_2/C_1 is 4 to 10 times smaller than that of quartz, giving wider ranges of resonance.

These synthetics can not replace quartz, however. Their temperature coefficients are at least 5 times, and in most cases 20 to 30 times, that of quartz.

One more piezo-electric material, of an entirely different type, has been developed. This material, barium titanate, is peculiar in that it is used in polycrystalline form. The normal piezo-electric effect is not noticed in a polycrystalline substance since the variously orientated crystals neutralize each other's action. Barium titanate, though, exhibits an "induced piezo-electric" action. This effect is derived from electro-striction, present in every substance.

In electro-striction, deformation X takes place, proportional to the square of the electric field,

$$X = k E^2$$

where k is a constant and E is field strength.

With an applied field E_a, a small

change ΔE causes an added deformation ΔX.

$$X + \Delta X = a\,(E_o + \Delta E)^2$$
$$\approx a\,E_o^2 + 2\,a\,E_o\,\Delta E$$
$$\Delta X = 2\,a\,E_o\,\Delta E$$

The additional deformation is proportional to the added field. This is equivalent to normal piezo-electric effect. With most substances this result is extremely small.

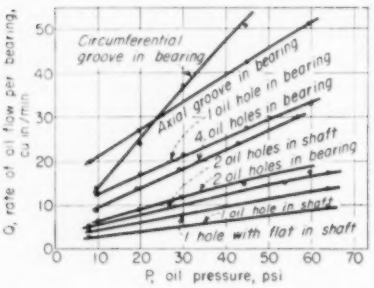
With barium titanate, on the other hand, the constant a is very large, millions of times greater than other substances. This material also remains polarized after the removal of the electrical impressment E_o, the external field. The advantage of barium titanate is that it is processed by ceramic means and can be given all sorts of shapes not possible with mono-crystals.

Oil Holes and Grooves in Plain Journal Bearings

Abstract from Technical Report 1449, National Bureau of Standards.

IN THE LUBRICATION of journal bearings, the method of admitting oil to the bearing constitutes an important problem in design. Fundamentally, the concept of the load-carrying film indicates that it is desirable to avoid oil holes or grooves which interfere with the normal development of hydrostatic pressure to support the load. In some bearing installations, however, it is not possible to satisfy this requirement.

Investigation covered five different arrangements for feeding oil through the bearing shell. These included one hole at the center of the unloaded side; two holes, one each on the loaded and unloaded side; four holes, each 45 degrees from the line of the load; one axial groove at the center of the unloaded side; and one circumferential groove. Tests were also made with three arrangements for feeding oil from the center of a hollow shaft—this is, one and two oil holes in the shaft, and a one-oil-hole arrangement terminating in a flat that extends along the surface of the shaft for one-half the bearing length. For all these arrangements each hole or groove was in the axial center of the bearing and was connected to the source of oil supply.


Q, rate of oil flow per bearing, cu in./min
P, oil pressure, psi
Circumferential groove in bearing
Axial groove in bearing
1 oil hole in bearing
4 oil holes in bearing
2 oil holes in shaft
2 oil holes in bearing
1 oil hole in shaft
1 hole with flat in shaft

OIL-FLOW versus oil-pressure curves for different feed arrangements when operating at a load of 3008 lb. per bearing and a speed of 2030 rpm. Q, the rate of oil flow per bearing in cubic inches per minute, is plotted against p, the corrected oil pressure in pounds per square inch.

ONE OIL HOLE IN BEARING. The absence of holes or grooves on the loaded side of the bearing permits normal development of pressure in the oil film. Consequently, the bearing has relatively low friction and low ZN/P at transition between stable and unstable lubrication. While the single hole does not allow the highest oil flow, it would probably provide adequate heat dissipation for most conditions.

TWO OIL HOLES IN BEARING. The single hole on the loaded side with this arrangement apparently disturbs the normal development of pressure in the oil film, causing a measurable increase in friction and in ZN/P at the transition point. The particular location of the oil hole on the unloaded side is adverse from the standpoint of oil flow; hence the rate of heat dissipation is somewhat lower than the arrangement with one hole in the bearing.

AXIAL GROOVE IN BEARING. This arrangement apparently provides a slightly better distribution of oil for the development of the load-carrying film than the arrangement with one hole in the bearing. Consequently, the bearing has low friction and lowest critical ZN/P, and the high oil flow provided is advantageous where forced cooling is necessary.

FOUR HOLES IN BEARING. The two holes on the loaded side disturb the development of pressure in the oil film to a greater extent than the arrangement with two holes in the bearing. Critical ZN/P is higher but the friction is comparable. The two holes on the unloaded side are not in advantageous positions, and its oil flow and rate of heat dissipation are comparable to the arrangement with one hole in the bearing.

CIRCUMFERENTIAL GROOVE IN BEARING. The groove dividing the bearing into two narrower parts increases the friction and causes a relatively high critical ZN/P. The high friction is counteracted by high oil flow and rate of heat dissipation.

ONE OIL HOLE IN SHAFT. The passage of one oil hole across the loaded portion of the oil film has little effect on both friction and critical ZN/P, which are only slightly higher than for the arrangement with one hole in the bearing. When the hole is exposed to some portions of the loaded area, the oil feed is practically shut off. Consequently, this arrangement results in the lowest oil flow and rate of heat dissipation.

TWO OIL HOLES IN SHAFT. With two oil holes for each journal, the disturbance to film-pressure development occurs twice in a revolution, resulting in slightly higher friction and critical ZN/P than with the arrangement of one hole in the shaft. The two holes also provide a greater oil flow and rate of heat dissipation. In this respect it is comparable to the arrangement with two holes in the bearing.

ONE OIL HOLE WITH FLAT IN SHAFT. The flat at the end of the oil hole causes considerable disturbance to the development of pressure in the oil film, increases the friction, and markedly increases the critical value of ZN/P. The oil flow and rate of heat dissipation are higher than in the arrangement with one hole in the shaft without the flat but not sufficient to counteract the effect of high ZN/P at transition from stable to unstable lubrication in the sleeve bearing film.

X-Ray Machine Grades Fruit

Abstracted from "Big Saving With New X-Ray Fruit Grader" by James Joseph, Food Industries, April, 1950.

A NEW FULLY AUTOMATIC X-RAY MACHINE grades and sorts oranges into six juice quantity classifications, detecting frost-damaged fruit at the same time. The principle is that oranges with more juice are less easily penetrated by x-rays. The juice collects at the center of frost-bitten fruit and is reabsorbed by the tree, leaving them relatively dry.

Oranges travelling in a cup conveyor are viewed by a photoelectric cell which adpusts the x-ray beam size to each orange. Larger fruit pass less radiation. The machine remembers the compensated beam size to be used with each orange.

The orange then passes under an x-ray unit where its opacity is compared to normal fruit. The electric current representing the difference factor enters a network which causes the orange to be pushed out of the conveyor into the correct bin. The x-raying and bin selection takes place in 1/100 second.

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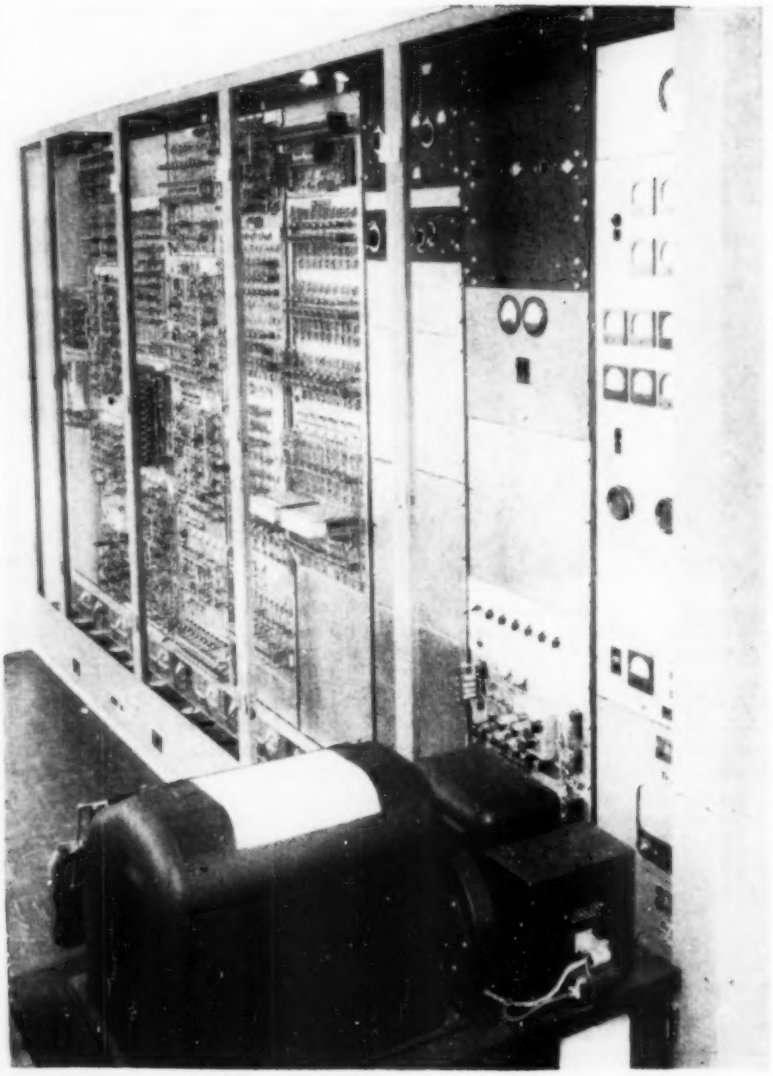


THE NATIONAL BUREAU OF STANDARDS SEAC computer is shown with operator's control table in foreground. The teletype keyboard and printer are used for direct input and output with numbers and instructions coded in hexadecimal notation.

Digital Computer is Fastest Yet Developed

RECENTLY COMPLETED BY THE NATIONAL BUREAU OF STANDARDS, the SEAC automatically sequenced electronic computer can handle seven basic orders faster than any other computer of this type.

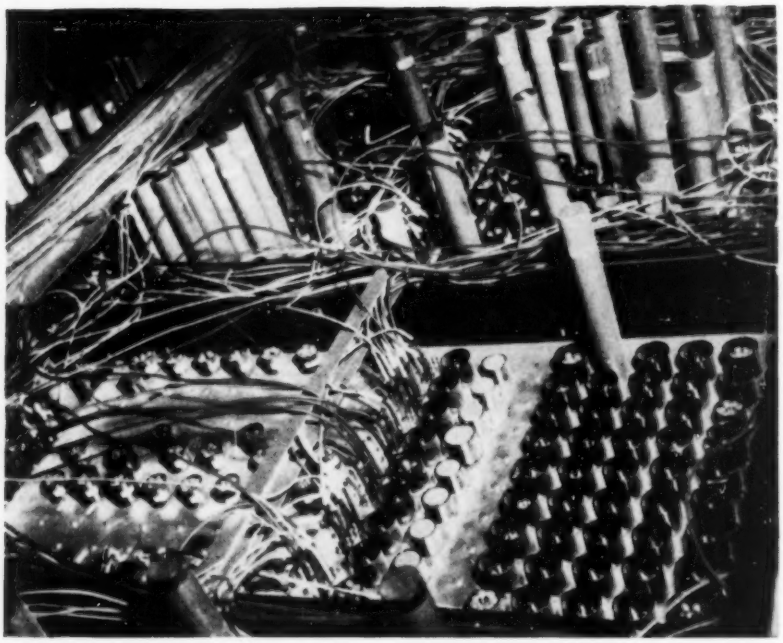
Because SEAC was needed for use as quickly as possible, its design was kept simple, and the equipment which had to be built was held to a minimum. It was designed to operate initially as a serial machine because this mode of operation requires less equipment than the parallel method; this does not, however, mean less functional effectiveness. The list of basic operations or orders was kept as short as possible for the same reason. SEAC has seven basic orders, chosen after a careful study which established their convenience for solving the most elaborate problems the machine is expected to handle: addition, subtraction, multiplication, division, comparison, logical transfer, and input-output control.

The comparison order allows two numbers to be compared to determine which is larger. This feature gives the machine a degree of choice. The result of a comparison thus dictates which of two alternative operational sequences the machine is to perform next. The logical transfer order makes it possible for the machine to alter an instruction it has previously received.

All operations are carried on in a binary number system, the presence of a pulse indicating "1" and the absence of a pulse indicating "0". In SEAC a sequence of 45 binary digits is known as a "word" and may convey operational instructions as well as numerical data. SEAC's high computing speed is largely the result of two design features: its rapid pulse rate (1 megacycle per second) and its large memory (512 words) capacity. They make it possible for SEAC to add or subtract pairs of 11-digit numbers 1100 times a second and multiply or divide them 630 times a second. These rates include the time it takes for the machine to search its memory for the numbers, operate upon them and return the result to the memory. An arithmetical operation of addition or subtraction alone is completed in 50 microseconds; multiplication or division is completed in 2500 microseconds.

SEAC's computing functions are carried on in four main sections: an input-output unit, a memory unit, a control unit, and an arithmetic unit. The input-output unit is the link between the machine and its operators. Here information and instructions are fed in and answers are printed out. The memory unit acts as a file for instructions and numbers while the control unit is directing the flow of information through the computer. Signals from the control unit perform the electronic switching which sends numbers and instructions to the right places at the right times. The control unit continually refers to the memory unit for instructions in the proper sequences, and the arithmetic unit carries out the actual computations the control unit directions to perform. When the arithmetic unit has arrived at the solution required, the control unit directs the memory unit to send the answer to the output unit.

The present input-output unit employs a manual keyboard for direct input and a teletype printer for direct output, using a hexadecimal notation (base 16) to represent numbers and instructions. Indirect operation is accomplished through the use of punched paper tape. SEAC's design allows for the later replacement of the teletype system by a faster magnetic wire system. The initial model of a



TYPICAL INTERIOR VIEW of the new computer shows the tube-base-mounted germanium-diode clusters used as high-speed electronic switches. The long fiber tubes enclose short lengths of continuously-wound electrical delay lines.

magnetic wire input-output unit with a single serial information channel has already been planned. The time to read a block of eight words in or out with the magnetic wire scheme is less than a tenth of a second; with the present punched tape it is 16 seconds. SEAC is flexible enough to accommodate a wide range of serial or parallel input-output systems, so that future developments can be directly incorporated.

All of these specific SEAC design considerations are a part of a much broader NBS program on automatic electronic computers. The program has four principal phases: fundamental research, engineering development, machine design and construction, and technical coordination.

The fundamental research includes basic studies in numerical analysis, logical design, programming techniques, and circuit efficiency. The long-range objective of this research is faster, simpler, and more versatile computers, and more effective use of the machines already in existence.

Ersatz Is Back

ACCELERATED INVESTIGATIONS leading to substitution of plastics, wood and other materials for critical and strategic metals and minerals is under way by several government departments.

The investigations are part of the stockpiling program. They form one segment of a larger program intended to strengthen U. S. resources. Increasing the efficiency of use of strategic and critical materials is another segment of the program. Operations are parcelled out among the 3 services, the Munitions Board, the Research and Development Board, and the Departments of Agriculture and Interior.

As part of the program the Army is reviewing specifications at the time of procurement with a view of substituting readily available non-critical and non-strategic materials. The Navy is running a systematic program to encourage redesign of essential parts of ships and components so as to use alternate materials, or, in some instances, to reduce the amount of critical material required per unit. The Air Force and the other services are saving material by procuring "miniaturized" components, particularly in the electronic field. The reduced size of parts thus scales down material requirements.

Masterminding the operations is the Munitions Board with the help of the Research and Development Board. The RDB has a 3-point program aimed at (1) finding alternate or substitute materials, (2) finding more effective ways of using critical materials, and (3) surveying developments which may reveal significant new requirements for wartime equipment. Operational responsibility for this "materials conservation" however, is in the Defense Department's three services. Here major savings of scarcest materials are possible through redesign and alternative specifications to enable shifts from scarcest materials to others.

An instance of this type of activity is the redesign of the Air Force jet engines where exacting requirements upon metals used within the turbojet has led to the use of extremely scarce alloying substances and a search is now under way to find more plentiful substitutes for these items.

Another example of "materials conservation" by substitution is in cast iron. Nodular cast iron, which is much superior to the older cast irons for many applications, has required the rare metal cerium for alloying purposes. Recent developments by the Navy and industry have resulted in a process for substituting the more plentiful metal magnesium as an alloying agent in the nodular cast iron.

Military researchers have found that asbestos and natural mica in electrical uses may be replaced with fibrous glass and synthetic mica. The Army is experimenting with nitriding as an anti-corrosion coating to replace such strategic materials as cobalt, chrome and molybdenum.

Pint-Sized Rocket Aids Experimenters

A ROCKET MOTOR small enough to hold in your fist is helping chemical engineers at the Massachusetts Institute of Techonology develop fuels for full-size rocket-powered missiles.

This tiny liquid-fuel rocket motor is believed to be the smallest of its type in the world. By running on very small amounts of fuel and by eliminating the need for elaborate safety precautions, it makes possible experiments costing far less than full-scale tests.

In research under the direction of Professors Hoyt C. Hottel and Glenn C. Williams of the M.I.T. chemical engineering department, the so-called "microrocket" has proved its value in testing rocket engine fuel efficiencies. The studies are under the immediate supervision of Bradford Darling and Saul Wolf of the M.I.T. research staff.

Already the results have brought about a more complete understanding

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of rocket fuel combustion. Future work will continue to fill in this picture and is also expected to provide answers to specific rocket design problems. The microrocket fuel testing is part of a comprehensive project now under way at M.I.T. under the auspices of the U. S. Navy Bureau of Ordnance and Office of Naval Research.

The M.I.T. microrocket uses only two pounds of fuel in the one minute during which 12 tons would be used in a rocket the size of a German V-2. The microrocket operates on exactly the same principle and with the same high efficiency as its larger prototypes —but on a greatly reduced scale.

Although built to use any liquid rocket fuels, the microrocket has to date been used chiefly with a combination of liquids which ignite spontaneously on contact. When they come together inside a rocket motor, the resulting flame makes a tremendous amount of heat and therefore power. Though a toy in size, the M.I.T. microrocket is hardly a living room plaything. On two pounds of fuel it runs for one minute, builds up more than 300 pounds of pressure inside the motor, shoots gas out its nozzle at a speed of about 5,000 mph, more than 2½ times the velocity of sound at these conditions, and produces heat at the same rate as does a furnace big enough to heat an eight-room house.

Pressed Metal Institute Meets In Cleveland

CLEVELAND, OHIO—Problems of metal stampers as component parts manufacturers, and the demands upon them during a period of mobilization will be major topics of discussion at the annual meeting of the Pressed Metal Institute, national association of metal stampers, September 11 through 17. The meeting will include a boat trip up the St. Lawrence and Saguenay Rivers with a three-day stopover at the Hotel Tadoussac located at the conflux of the rivers.

This PMI convention will be unusual for there will be no speakers and

Lehigh's Second Product Design Seminar Completed

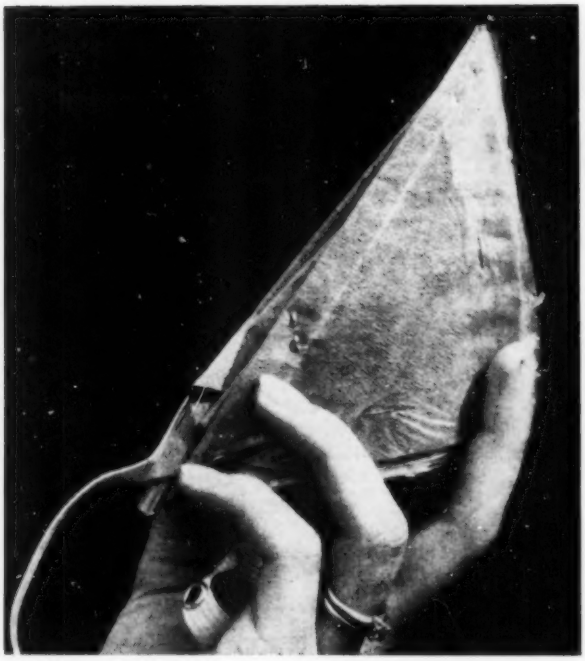
MOVING FROM A STAGE described as a "shake-down cruise" to that of well-defined service, Lehigh University's Product Design Seminar in Bethlehem, Pa., has completed its second year with 43 delegates from 15 industrial firms participating.

Commenting that "it has now become well recognized that we are not running a summer school for designers," Raymond Spilman, co-director for the Society of Industrial Designers which jointly sponsors the Seminar, stressed that the program is drawing top level representatives of management and engineering.

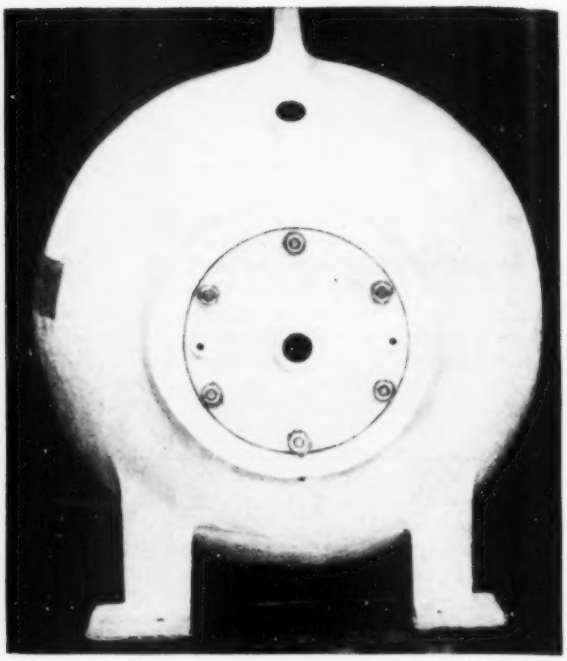
One of the major objectives of the Seminar, according to Spilman and Einar M. Ramberg, Lehigh University co-director, is to carry the philosophy of unity throughout the development of the product. "The idea is to bring product development out into the open and have all representatives of design, engineering, and management involved in the discussions right from the start," Spilman said, adding that "for the protection of the investment of management, the engineer and designer should sit at the management level during the development of the product."

As in the initial year, delegates to the Product Design Seminar heard talks and held discussions with 38 design authorities including Egmont Arens, Walter Teague, Harold Van Doren and Gyorgy Kepes. As an innovation, industrial editors were included in the program with George Nordenholt, editor of PRODUCT ENGINEERING joining in a discussion of the coordination of design and engineering.

Seminar delegates watch as PE's editor George F. Nordenholt and Industrial Designer Peter Müller-Monk examine a new product.



A DISPOSABLE CONSTANT-FLOW OXYGEN MASK is being produced for emergency use on airlines and non-military passengers on military lines. It will cost about 25 cents when it gets into mass production. As the wearer breathes in oxygen, the bellows expands and contracts and the gases he exhales escape through the porous paper. The disposable mask is a development of the Air Material Command's Aero Medical Lab at Wright-Patterson Air Force Base, in conjunction with the H. L. Burns Co. of Portland, Oregon.



Steel Founders' Society of America

GIANT CAST STEEL DIVING BELL, known as a benthoscope, in which Otis Barton, Boston scientist, soon will undertake mile-deep undersea explorations in Pacific waters off La. Jolla, Calif. The massive cast steel bell is designed to withstand pressures beyond 3,000 psi with assured safety at depts of at least 6,000 ft and theoretical safety factors down to 10,000 ft. Cast in an alloy steel, the benthoscope was made by Atlas Steel Casting Company, Buffalo. Taller than a man, it weighed 5,400 lb without equipment installations, and has machined walls 1¾ in. thick.

no speeches. Instead, a continuous series of committee meetings, both on the boat and at the hotel, has been planned. Each of these will be concerned with problems resulting from current international conditions.

When in session personnel of the committees will be augumented by Institute members having direct or indirect interests or special skills which might further the primary objectives of some phase of a committee's current activity.

Emphasis also will be placed upon the completion of plans for mobilizing the facilities of the stamping industry and for establishing basic manpower requirements.

British Extrusion-Coat Aluminum Electrodes

LONDON—A British firm, Murex Welding Processes, Ltd., has recently announced the production of aluminum welding electrodes with an extrusion-coated flux. This change from dip-coating to extrusion coating has long been considered a possibility by American electrode manufacturers, but up until this time no U. S. firms have seen fit to go ahead with the necessary development work required.

The new electrode is being produced to a new formula which, it is claimed, gives it several useful characteristics. One of the most important of these is that the running properties are comparatively smooth, the spatter often associated with aluminum welding being greatly reduced and the arc easier to control. Consequently the electrode can be used in any position, even overhead, although it is essentially designed for welding in the downhand position.

Another advantage which has been achieved in the development of this new electrode is that the slag is easy to remove. On a close butt weld on suitable plate the slag is self-releasing. Even when this result is not obtained the slag will be far easier to remove than that produced by previous types of electrodes.

The coating of the new Murex aluminum-silicon extruded type differs entirely from the dipped coating; it is more stable in storage and is much less likely to pick up moisture. Nevertheless, like all aluminum types, these electrodes should be stored in a dry place to avoid the possibility of moisture absorption.

ASME Meeting Features Textiles

THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS will hold its Fall meeting in Worcester, Mass., Sept. 19-21 at the Hotel Sheraton. Some 50 papers in such fields as heat transfer, management, rubber and plastics, textile, power, machine design, safety, gas turbine power, fuels, hydraulics, materials handling, production engineering, metals engineering and wood

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A "TELEPHONE" DIAL on a dispatcher's desk in the Shell Oil Company's New York City office now operates, by remote control, four new, G-E equipped, automatic pumping stations on the oil company's eight-inch petroleum products pipe line almost 1,000 miles away. To reach any of the four special controlled pump stations along the line, dispatchers merely dial code numbers. The dialing activates equipment at the stations to start or stop pumps or request information on operating conditions. A unique telemetering system then automatically sends reports back to New York. The new system also enables the dispatcher to sound a siren at each station to summon the man on duty to handle any emergency that can't be controlled by the new remote control system.

industries will be delivered at 24 technical sessions.

Appropriate to the New England locale, there will be four textile sessions. Topics to be covered at these sessions include plastics in the textile industry, industrial applications for nylon plastics, basic distribution of cost dollar in textile manufacturing, site selection and construction of worsted and woolen mills for low cost operation, production crimping of natural fibers and the construction of modern cotton and rayon mills.

Plastic Helmets For Army

SOLDIERS FIGHTING IN THE FAR EAST may be issued a new type of plastic helmet. It is made from compressed nylon fiber. The Army is procuring a large amount of them for field test purposes. Army Ordnance spokesmen say that the new helmet has greater bullet protection than the old steel helmet and also weighs less. Earlier this year Army spokesmen told a Congressional Committee that the new type helmets are more expensive than steel, but that with increased production they should cost less. The material itself is cheaper than steel.

"Dry" Bearings Show Promise

RECENT TESTS of hot-pressed silver bearing materials at the U. S. Flight Propulsion Laboratory, Cleveland, Ohio, are expected to lead to practical applications of a tri-metal alloy.

Experiments were conducted by 3 laboratory researchers to determine the feasibility of lubrication by transfer of molybdenum disulfide (MoS_2) from the structure of a composite bearing material to the contacting surface of sliders at sliding velocities of 75 to 8,000 fpm and loads of 269 to 1,017 grams. Loaded cylindrical specimens had hemispherical surfaces contacting rotating steel disks. This method of lubrication produced an effective surface film with materials containing more than 5 percent molybdenum disulfide, 5 percent copper and silver. The best composition with consideration for both friction and wear contained 10 percent MoS_2, 5 percent Cu and 85 percent Ag.

A copy of the 22-page report of experiments is available from the Library of Congress, Washington 25, D. C. at $2 in microfilm or $3.75 in photostat form.

A BERLIN ENGINEER will soon complete an amphibious car with which he hopes to drive directly from Germany to New York. He plans to cross the Atlantic in about two weeks and then mass produce his car—if he lives.

•

THE PATENT OFFICE, after long and grave consideration, issued patent No. 2,513,494 to Albert J. Kline, Douglas, Ill., for his cowtail holder.

•

RUBBER, steel, and oil will be the first U. S. strategic resources to be affected by the present national defense speed-up.

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THE NATION'S NOISE EXPERTS will meet on October 20 at the Armour Research Foundation of Illinois Institute of Technology to discuss excess decibels.

•

GERMAN PATENT OFFICE is back in business. Its recent issue of 12 patents was the first activity since 1945. The majority of patents went to the coal mining industry.

•

BUICK CARS now have an optional safety plate glass that cuts sun glare and reduces heat transfer. It has a slight bluish-green tint to reduce glare.

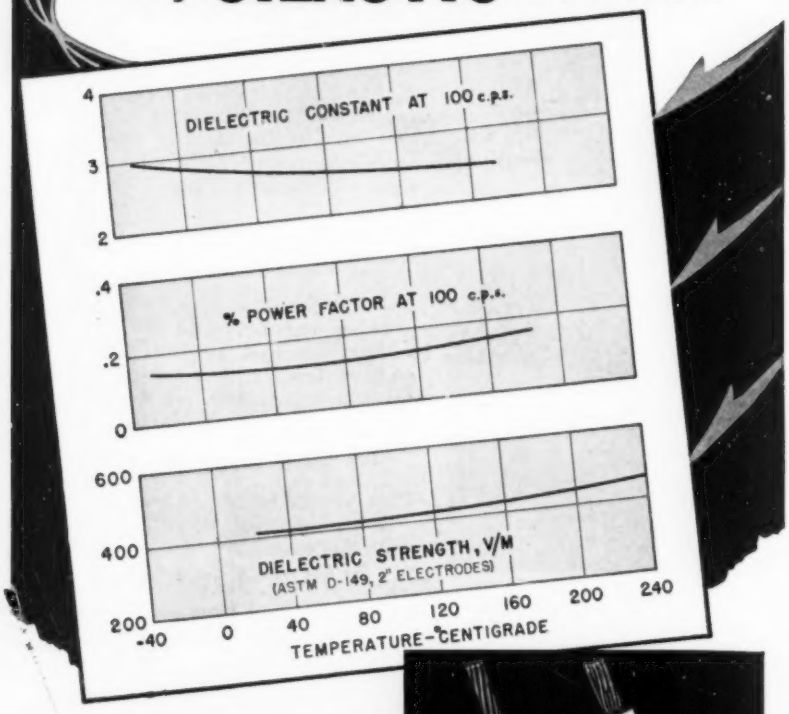
•

THE ATOMIC ENERGY COMMISSION has selected the E. I. du Pont de Nemours Co. to build and operate a huge new group of atomic production facilities.

•

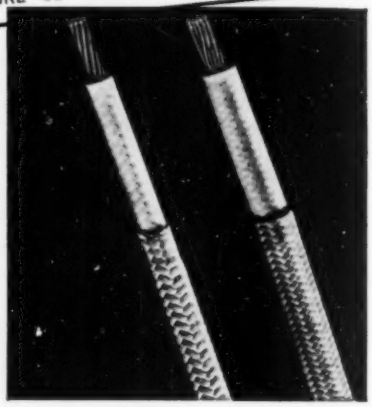
PURDUE UNIVERSITY will be host to the American Foundrymen's Society on Nov. 2 and 3. The occasion will be the third annual Metals Casting Conference.

why SILASTIC* works best



as a dielectric material from −100° to +500°F.

Over that wide temperature span only Silastic, the Dow Corning Silicone Rubber, remains resilient and retains high resistance to weathering, moisture, oxidation and ozone. Add good dielectric properties to those advantages and you have the reasons why Silastic is an excellent insulating material for high temperature, high voltage cable and for use in equipment where mechanical breakdown limits the effectiveness of resinous insulating materials.



Silastic is extruded over wire and cable ranging in size from No. 22 to 500,000 circular mils to provide insulation at temperatures from −80° to 400° F.*

Many engineers are familiar with Silastic as a remarkably heat stable and oil resistant rubberlike gasketing and sealing material for use between −100 to 500° F. Silastic as a dielectric for ignition and intercommunication cable and for field and armature coils is, however, a relatively new development. That's why Dow Corning has made available reprints of a recent article giving the most up-to-date information on the physical, chemical and dielectric properties of Silastic. To receive your copy, phone the nearest branch office or write for Reprint No. R-21A.

*T.M. REG. U.S. PAT. OFF.

DOW CORNING CORPORATION
MIDLAND, MICHIGAN
Atlanta • Chicago • Cleveland • Dallas
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Dow Corning
FIRST IN SILICONES

SILICONE NEWS

THAT'S SILICONE SERVICE TO INDUSTRY

Cost conscious maintenance men throughout industry now specify Class H Insulation made with Dow Corning Silicones. It keeps hard working, critical motors on the job. Here's a typical example from the steel industry. A cupola crane hoist motor insulated with the best Class B materials failed 22 times in 1098 days. Average life was only 50 days; rewind costs alone amounted to $3,634.

Then National Electric Coil Company of Columbus, Ohio, rewound the motor with Class H (DC Silicone bonded and impregnated) Insulation at an extra cost of only $79. That motor, still in good condition after 613 days on the hoist, was transferred to the trolley bridge. It's still in service after a total of 1521 days. That's silicone service to industry.

HERE'S WHAT IT MEANS TO YOU

Dow Corning Silicone Insulation offers greater freedom in the design of electrical machines. With this remarkably heat stable insulation you can reduce the size and weight of motors, generators, transformers and solenoids by 50 to 100%. That's because Dow Corning Silicone Insulation withstands temperatures 100° C. above Class B limits.

For more information on how to lower maintenance costs, increase productivity and improve electrical machine design with Silicone (Class H) Insulation, phone our nearest branch office or write Dept. R-21.

REMEMBER

Dow Corning Silicones Mean Business

UNIVERSITY OF MICHIGAN LIBRARIES

Spur Gear Durability

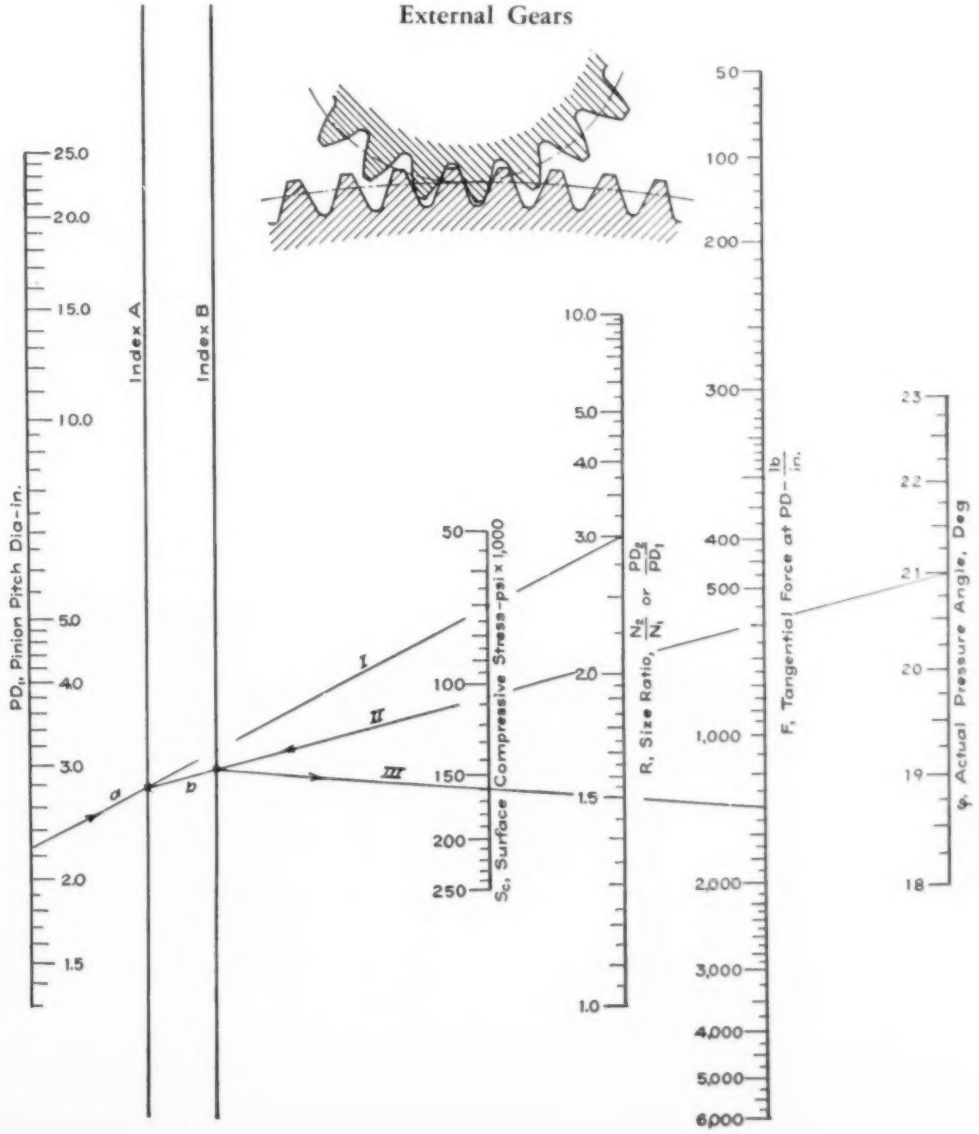
J. E. RICE
Gear Engineer, Wright Aeronautical Corp.
Woodridge, N. J.

CALCULATION OF THE LIMITING LOAD for wear of steel gears can be simplified by these nomograms. Charts are based on the Hertz equation for compressive stress of curved tooth surfaces at the pitch diameter. The most important factors used in designing gear drives are conveniently expressed in this chart, such as: the number of teeth, pitch diameters, and the actual pressure angle. The modified Hertz equation for external, steel gears is given by:

$$S_c = 4580 \sqrt{\frac{F\,(N_1 + N_2)}{N_2 \cdot PD_1 \cdot \sin 2\phi}}$$

where: S_c is the compressive stress in psi; F is the tangential force at the pitch diameter in lb./in.; N_1 and N_2 are the number of teeth in the pinion and gear respectively; PD_1 is the pinion pitch diameter in inches; and ϕ is the actual pressure angle in degrees. For internal gears, the

External Gears



What a Business Machine Executive learned about DIE CASTING from an Automotive Specialist...

A NOTED business machine executive states there are approximately 1,000 parts in the average adding machine and each of these parts creates its own special production problem. Just recently one of the larger automotive suppliers showed him how, through die cast parts, a considerable saving in the production of these small parts was possible. Today, this saving permits him to place his machine on the market at a more favorable price, an important factor for every manufacturer in the buyer's market.

If you have a die casting requirement in which quality or price is a problem, you are invited to write to

THE ELECTRIC AUTO-LITE COMPANY
Die Casting Division

Woodstock, Illinois
600 So. Michigan Ave., Chicago 5, Illinois

Lockland, Ohio
723 New Center Bldg., Detroit 2, Michigan

AUTO-LITE

die castings
plastics
wire and cable
industrial
thermometers

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plus sign should be changed to minus.

Extensive testing of aircraft gear drives has determined an allowable compressive stress value of 160,000 psi. This applies to tooth surfaces that have been hardened to Rockwell C 60. For steels with a lower hardness, the allowable compressive stress should be reduced proportionately with the hardness to a value of 75,000 psi at R_c 20.

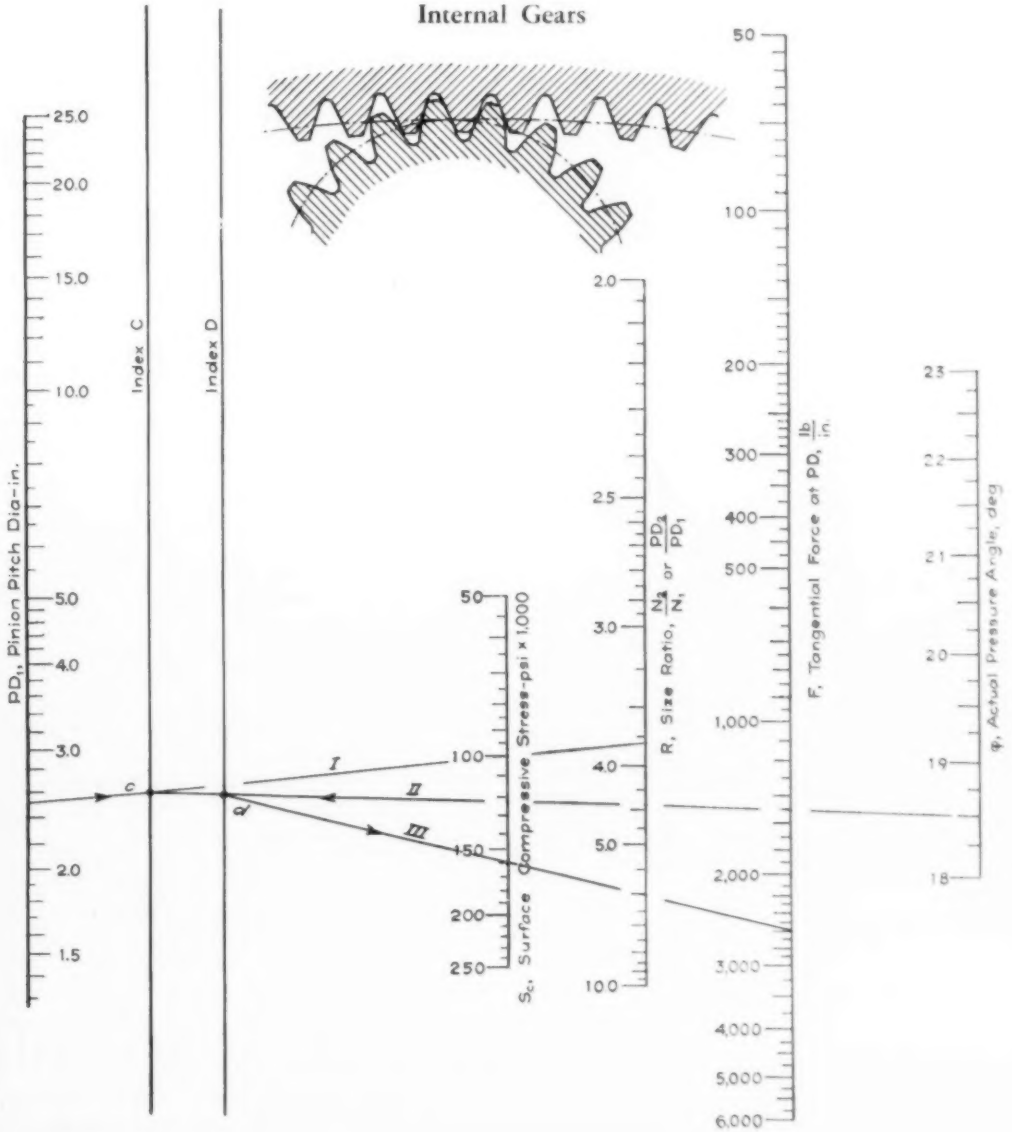
EXAMPLE: Find the minimum width of an external gear drive to transmit a 1,500 lb. load. The gear and pinion are 6.75 and 2.25 inches respectively, while the actual pressure angle is 21 degrees.

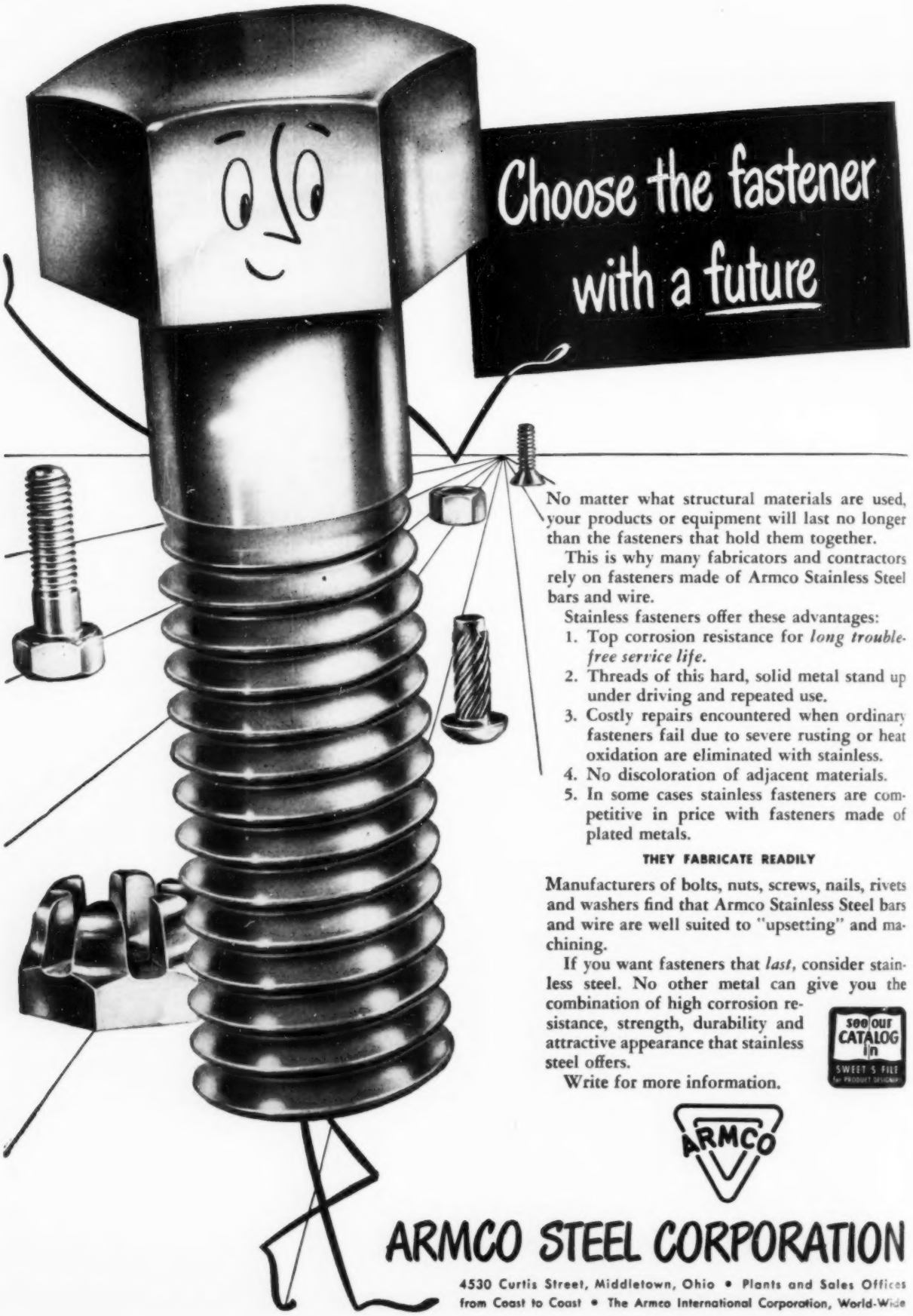
Line I is drawn between $PD_1 = 2.25$ and the size ratio, $R = PD_2/PD_1 = 6.75/2.25 = 3.0$ to Index Scale *A* to establish point *a*. A line (II) drawn between point *a* and the pressure angle of 21 deg. determines point *b* on Index Scale *B*. Projecting line III from point *b* through the maximum stress $S_c = 160{,}000$, gives the allowable tangential force of 1,400 lb/in. Using this value, the minimum tooth width would be 1,500/1,400 = 1.07 inches.

EXAMPLE: Determine the minimum tooth width of an internal gear drive which is to transmit a force of 2,000 lb. The pitch diameters are: pinion is 2.5 in., and internal gear is 10.5 in., and the pressure angle is 18.5 degrees.

PROCEDURE: Draw line I from $PD_1 = 2.5$ to size Ratio $R = 10.5/2.5 = 4.2$ to obtain point *c* on the Index Scale *C*. Line II is drawn between point *c* and the 18.5 deg pressure angle. Point *d* is determined by the intersection of this Line II and the Index Scale *D*. From point *d*, line III is projected through the limiting stress of 160,000 psi to the tangential force of 2,600 lb/in. Minimum width of the teeth is thus found to be 2,000/2,600 = 0.77 inches.

Internal Gears





No matter what structural materials are used, your products or equipment will last no longer than the fasteners that hold them together.

This is why many fabricators and contractors rely on fasteners made of Armco Stainless Steel bars and wire.

Stainless fasteners offer these advantages:

1. Top corrosion resistance for *long trouble-free service life.*
2. Threads of this hard, solid metal stand up under driving and repeated use.
3. Costly repairs encountered when ordinary fasteners fail due to severe rusting or heat oxidation are eliminated with stainless.
4. No discoloration of adjacent materials.
5. In some cases stainless fasteners are competitive in price with fasteners made of plated metals.

THEY FABRICATE READILY

Manufacturers of bolts, nuts, screws, nails, rivets and washers find that Armco Stainless Steel bars and wire are well suited to "upsetting" and machining.

If you want fasteners that *last*, consider stainless steel. No other metal can give you the combination of high corrosion resistance, strength, durability and attractive appearance that stainless steel offers.

Write for more information.

See our CATALOG in SWEET'S FILE for PRODUCT DESIGNERS

ARMCO

ARMCO STEEL CORPORATION

4530 Curtis Street, Middletown, Ohio • Plants and Sales Offices from Coast to Coast • The Armco International Corporation, World-Wide

Nomograph for Weight of Metal Bars

CARL P. NACHOD
Vice President,
Nachod and U. S. Signal Co., Inc.

THIS NOMOGRAPH provides a convenient and simple means of determining the weight of metal bars which may be either circular, square or hexagonal in shape. Chart has been constructed to include the densities of several different metals.

To use this chart:

1. A line (I) is projected horizontally from point *r* on the A Scale to point *s* on the B Scale. The point *r* corresponds to the cross-sectional dimension, *D*, of the bar.

2. Draw a line (II) from point *t* on the Lower C Scale (which defines the shape of the bar) through point *s* and extend it to intersect the A Scale at turning point *u*.

3. A line (III) is drawn from turning point *u* to point *v* on the Upper C Scale which identifies the proper metal.

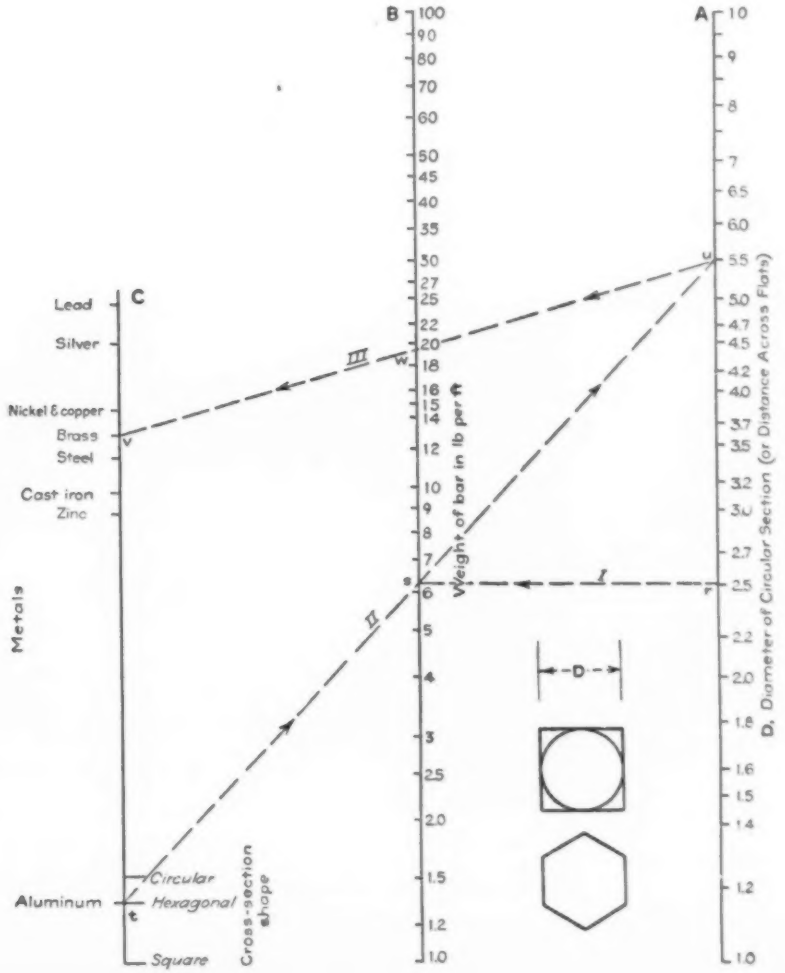
4. The intersection of line III on the B Scale gives the weight of the metal bar in pounds per foot.

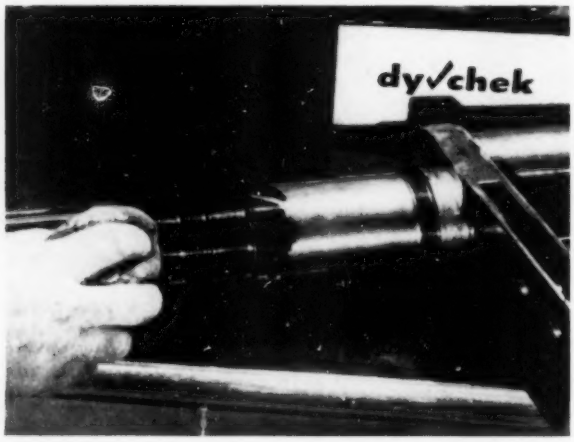
EXAMPLE: Determine the weight of a brass Hexagon bar of 2.5 in. (across flats).

At 2.5 in. on the A Scale a horizontal line (I) is drawn to intersect the B Scale. Using this point and choosing the hexagonal point on the Lower C Scale, a line (II) is projected to the A Scale. A line (III) is then drawn between this point and the brass metal on the Upper C Scale. The intersection of line III and the B Scale provides the weight of 19.4 pounds per foot.

By increasing the Metals Scale C, this nomograph can be used for other materials. This is done by first calculating the weight of 12 cubic inches of the material in question. Then, the numerical value of this weight on the A Scale is horizontally projected to establish the proper point on the C Scale.

The weight of hollow bars can be determined from this chart by means of subtraction. It is to be noted that this chart is applicable for bars with a cross-sectional dimension of from 1 to 10 inches.





RED DYE is brushed or sprayed on after surface has been washed with a special cleaning fluid. It is allowed to penetrate for 5 to 15 minutes and is then rubbed off as shown. Cloth takes dye out of threads and normal surface indentations. Only in tight crevices is the dye retained; below surface and invisible.

WHITE DEVELOPER is applied after surface has again been washed with volatile cleaner. As developer dries it draws red dye out of cracks to show bright red flaw lines. The process works on magnetic and non-magnetic materials. The fluids are harmless to hands or metals. Internal cavities are not shown.

Cracks in Metal Parts Show Up With Red Dye

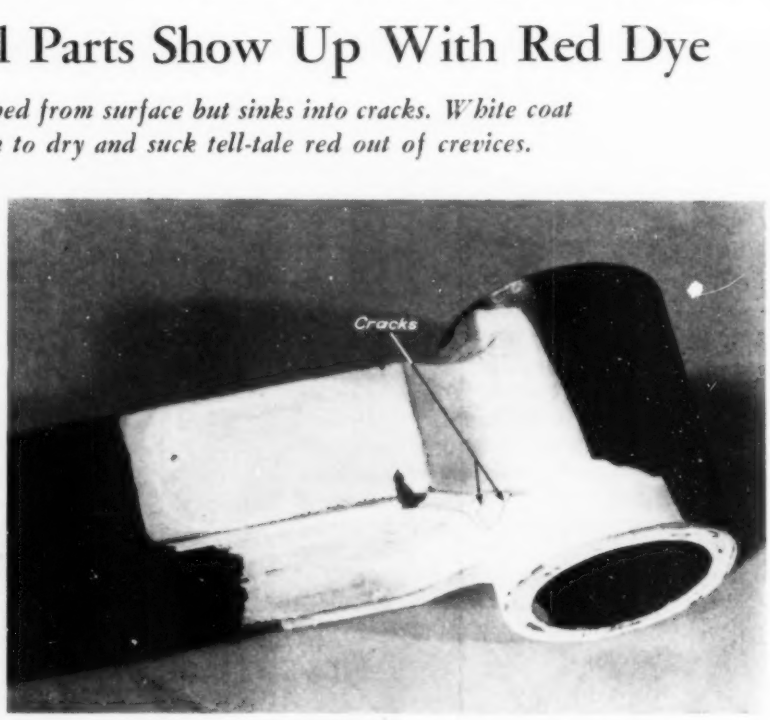
Red liquid is wiped from surface but sinks into cracks. White coat is painted on to dry and suck tell-tale red out of crevices.

A metal inspection process has been developed to detect surface cracks in metal parts. Disassembly of parts is usually not necessary. No auxiliary equipment is needed. The process was first used in the aircraft industry on gas turbine parts.

The surface is cleaned and washed with a special cleaning fluid. A red dye, with very low surface tension, is then applied. This red dye works deep into the slightest surface crack. After 5 to 15 minutes the surface is wiped and cleaned. A white developer is put on. This developer dries and draws the red dye out of deep cracks. Cracks show as brilliant red lines on white developer.

The process works on magnetic or non-magnetic metals. The fluids are non-corrosive and non-toxic. One man does the job, lasting 20 minutes in all. Strictly internal flaws of course cannot be detected. There must be a crack.

Dy-Chek Co., Div. of Northrop Aircraft, Inc., 1515 E. Broadway, Hawthorne, Calif.



CRACKS SHOW UP as red lines. Faults that are invisible to ordinary inspection, even with a magnifying glass, are spotted easily. High stress parts can be inspected without disassembling the whole machine. Welds are checked easily.

Circle No. 1 on Reader Service Card

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Electric Clutch Holds Synchronous Speed Without Slip

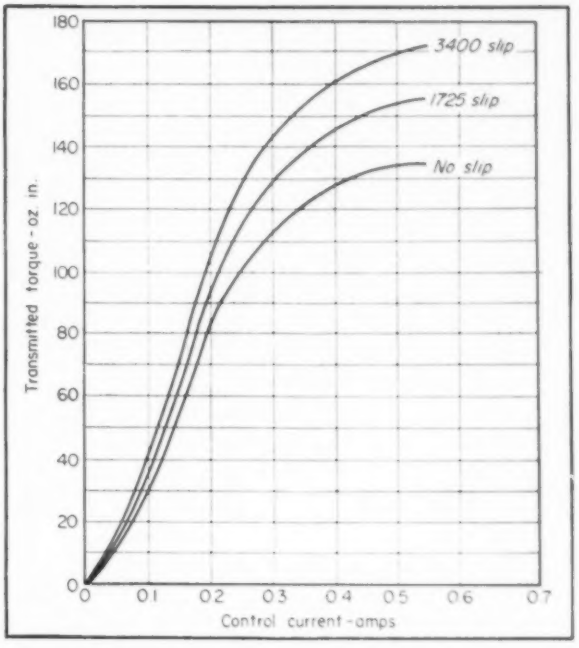
Also operates with slip, giving increased torque from eddy-current effect.

These fractional hp electric clutches are capable of synchronous operation, at zero slip. Capacities up to ½ hp are now available, with 30 watt control power.

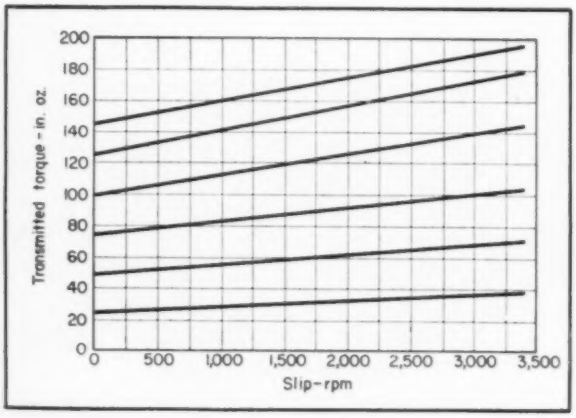
For illustration, the driven member can be thought of as a disk of high-hysteresis, permanent-magnet material. The driving members are two saw-edge cookie cutters, sandwiching the driven disk and rotating on the same axis. Each tooth is an electro-magnet, fed by the control current. The disk is cut by flux lines and pulled along as the driving members rotate. A given control current with its resultant flux density will hold the disk tightly, without slip, up to a certain torque load. As this maximum synchronous torque is exceeded, slip begins. Torque transmitted magnetically remains constant, while slip induces eddy currents in the disk to give an additional torque-carrying factor. Transmitted torque increases as slip and its resultant eddy currents increase.

A larger control current gives heavier flux density and allows higher torque to be transmitted. There is no mechanical adjustment in the clutch. Slip speeds above 3,000 rpm are handled. The new clutch is claimed to be more independent of heat losses than convention fluid-magnetic, fluid or friction clutches. There is no mechanical variation of the air gap. The flux itself is varied instead.

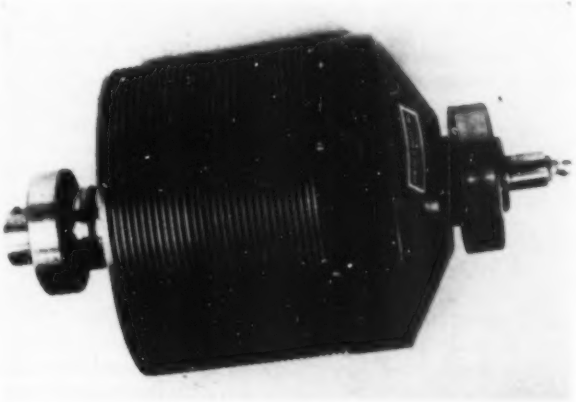
Duncan & Bayley, Inc., 785 Hertel Ave., Buffalo 7, N. Y.



MAXIMUM TORQUE TRANSMITTED without slip increases with applied current. Greater torques bring slip. Advantages of magnetic and eddy-current clutches are combined. Eddy current torque varies from 3 to 10 percent of net torque.



RATED TORQUE is transmitted without slip by magnetic attraction. Slip, with eddy currents gives increased torque. Each curve represents a set control current. Allowable continuous torque at any slip rpm is of course limited by temperature.



MAGNETIC CLUTCHES up to ½ hp have already been made. Torque transmission capacity at constant control current is not effected by slip-heat loss. Overall shaft length on the ½ Hp model is 7½ in., body length 4¼ in. and outside diameter 4 inches.

Circle No. 2 on Reader Service Card

CONTINUED ON PAGE 166

Amplifier Used With Strain Recording Oscillograph

This amplifier, when used with a magnetic oscillograph, provides a complete unit for the measurement of strain or other phenomenon where a resistance sensitive pickup is employed. This equipment records static or dynamic strains up to 100 cps; direction and magnitude of the strain can be read from the chart.

Four active gages may be used. When less are employed, dummy gages or precision resistors must be substituted to complete the bridge network. One of the dummy gages may be used for temperature compensation when a single active gage is employed. An internal calibrating resistor may be connected in the bridge circuit to adjust overall gain. The calibrating resistor is intended for the SR-4 120 ohm strain gage but may be changed for the calibration of individual gages, so that strain can be read directly in micro-inches per inch from chart deflections.

Although primarily intended for use with the SR-4 120 ohm strain gage, it is possible to use any gage type with ohmic resistance of 50 to 1,000 ohms. The instrument can be used for recording pressures, temperatures, accelerations or forces, provided the equipment is calibrated in terms of the particular pickup used.

Terminals are provided for connection of a bridge switch to permit sequential readings of six pre-calibrated strain gage circuits of four gages each. The output of the amplifier consists of three stages, direct coupled, designed to match the characteristics of the direct inking oscillograph. This section may be used as a general purpose, low gain d-c amplifier with gain and centering controls provided. The amplifier may also be adapted for use with graphic recorders. Power required is 135w, 100 to 125 v, 60 cycles.

Overall dimensions are 17¼ x 11 x 7½ in., weight is 30 lb. Recommended gages are 120 ohm type, but 50 to 1,000 ohm gages may be used. Output impedance is 50 ohms. The amplifier must not be loaded with less than 150 ohms. 12 tubes are required in all.

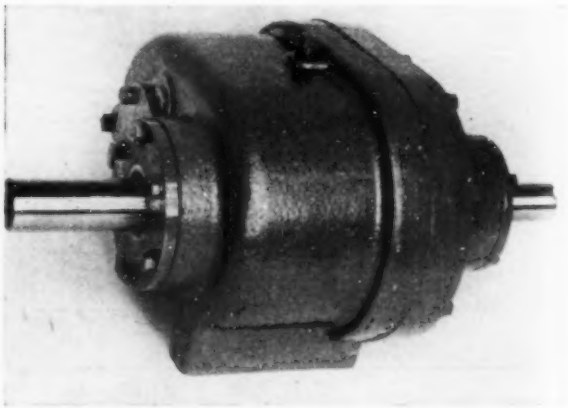
Brush Development Co., 3405 Perkins Ave., Cleveland 14, Ohio

Circle No. 3 on Reader Service Card

Double Clutch Transmission Has Two Output Speeds

A two-speed automatic transmission of 20 hp capacity, with minimum input speed of 900 rpm, has been announced. The mechanism consists of two over-running clutches, one engaging in a clockwise and the other in a counter-clockwise direction, so arranged that by reversing the motor, two speeds are provided, one at direct motor speed and the other at a pre-selected reduction, without changing direction of rotation of output shaft. -Motor controls are accomplished by means of a push button, limit, or timing switch or other electrical means. In the 20 hp model, the low limit of reduction is 6 to 1, but any pre-selected ratio, from direct motor speed to this low limit is obtainable. Other models are built to handle 5 to 50 hp motors. The transmission has general application to machine tool controls, and is also adaptable to special machinery in chemical process and service industries.

Constant speed motor drives make practical more selective speed ratios and multi-speed motors make a proportionately greater range of speed changes available.



Western Mfg. Co., 3400 Scotten Ave., Detroit 10, Mich.

Circle No. 4 on Reader Service Card

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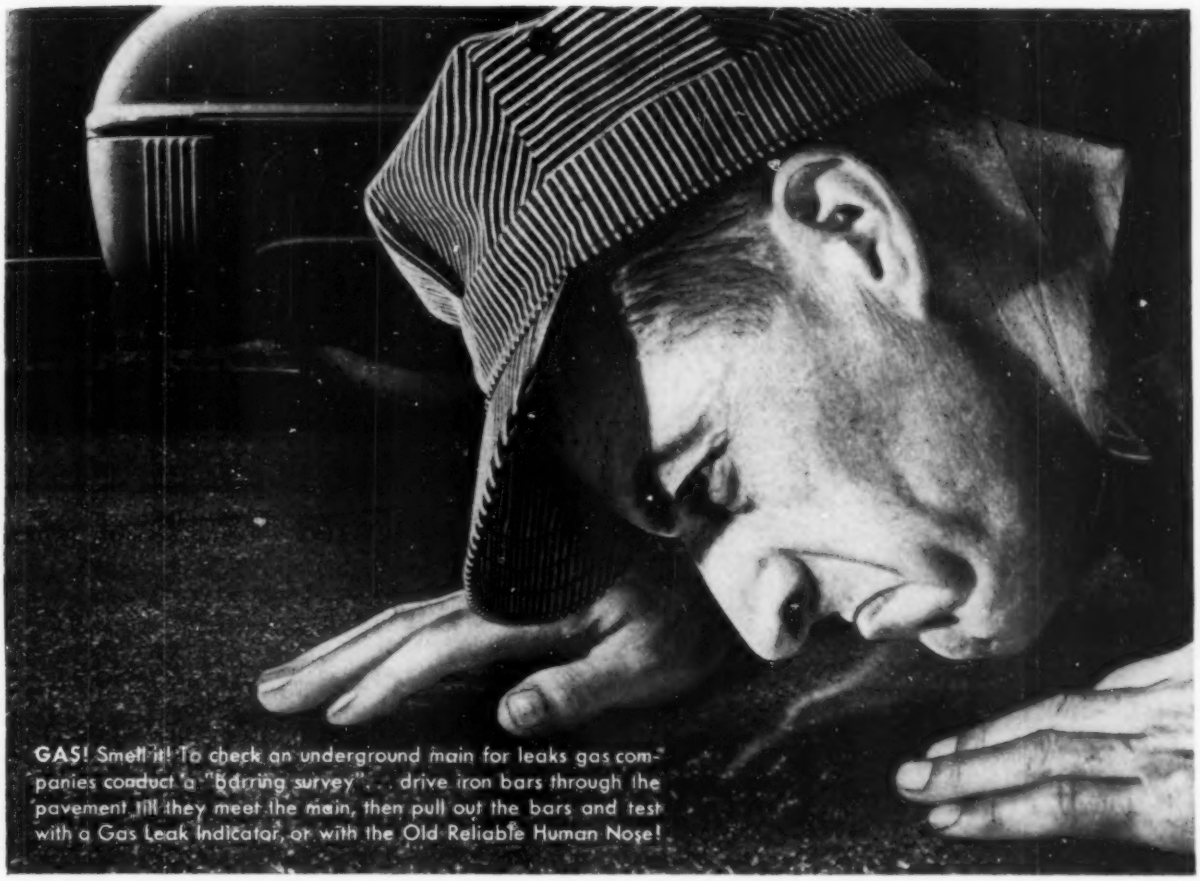
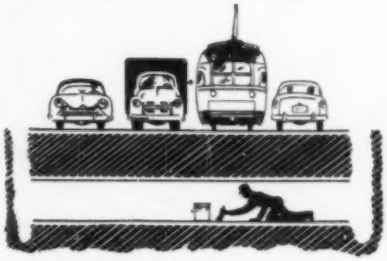


GAS! Smell it! To check an underground main for leaks gas companies conduct a "barring survey" . . . drive iron bars through the pavement till they meet the main, then pull out the bars and test with a Gas Leak Indicator, or with the Old Reliable Human Nose!

3M Sealer smothers $50,000 smell for only $7,000

When a 30-inch gas main springs leaks under one of the busiest traffic thoroughfares in town, your nose knows it . . . and the gas company knows fast action is necessary.

Naturally the western gas company that owned this leaky main was worried. A barring survey disclosed



50% to 100% gas indications over a 356-foot section of piping . . . and a cost survey proved that repairing it the orthodox way would require more than $50,000, including expense of temporary trackage for a parallel streetcar line and an intersecting railway spur.

The answer? Separate shaft excavations to plug the pipe at each end . . . plus application of 3M Sealer EC-801 by one man *inside* the main. Pre-testing proved this 3M Sealer resistant to internal and external pressures, and all corrosives, solvents and contaminants likely to be encountered. Total cost . . . less than $7,000.

Worth remembering: there are 3M Sealers, Coatings and Adhesives especially compounded to meet the most unusual requirements of almost every industry . . . to withstand the punishment of almost every kind of corrosion and weathering . . . to last enduringly, protect completely and hold tight on critical jobs of every kind. And if we're not making the one you require to meet your specific problem, a letter to us will have our engineers on the job in a hurry.

Call or write . . . no obligation. And ask for your copy of "3M Adhesives • Coatings • Sealers."

3M BRAND
ADHESIVES-COATINGS
SEALERS

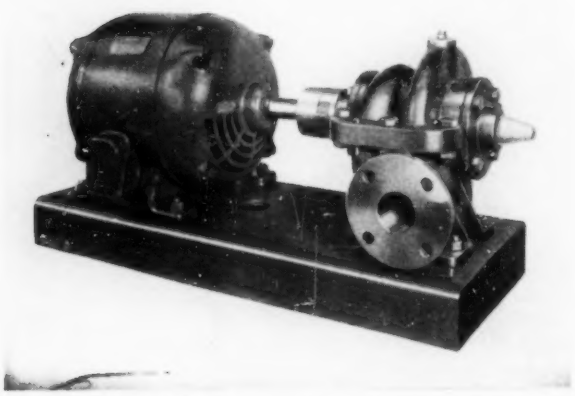
Made in U.S.A. by Minnesota Mining & Mfg. Co., Adhesives And Coatings Div., 411 Piquette Ave., Detroit 2, Mich. Also makers of "Scotch" Brand Pressure-sensitive Tapes, "Scotch" Sound Recording Tape, "Underseal" Rubberized Coating, "Scotchlite" Reflective Sheeting, "Safety-Walk" Non-Slip Surfacing, "3M" Abrasives. General Offices: Saint Paul 6, Minnesota. General Export: Durex Abrasives Corp., New Rochelle, N. Y. In Canada: Canadian Durex Abrasives, Ltd., Brantford, Ontario.

All Wearing Pump Parts Removed in One Assembly

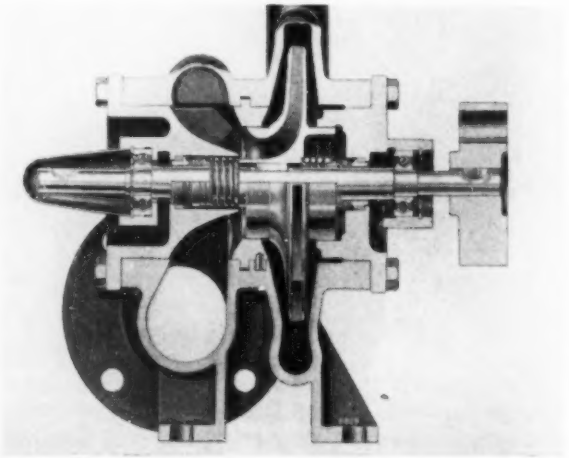
To avoid replacement of individual bearings, wearing rings and other parts, a pump has been designed in which every wearing part is contained in one removable rotor assembly. The entire unit can be lifted out and replaced. Three capacities of the centrifugal pumps are built with 450 gpm capacities and 230 ft heads as maximum.

Mechanical seals are used in place of stuffing boxes. Pump bearings are lubricated for life. The cast iron case is simpler without the usual cast-in brackets to hold bearings. Shaft is stainless steel. It is extended on both ends to permit motor coupling to either end. One ball bearing is used as a locator bearing. Impeller is precision cast bronze, shrunk and keyed to the shaft.

De Laval Steam Turbine Co., Trenton 2, N. J.



FOR MAINTENANCE, 1) remove the top cover and end plate studs, 2) lift off the top cover, 3) lift out assembly of end plates, shaft, bearings, seals and rotor.



MECHANICAL SEALS are used on shaft. There are no stuffing boxes to be repacked. Ball bearings in both pump and motor are lubricated for life. Motor is coupled to either end.

Circle No. 5 on Reader Service Card

Gear Motors Operate Single or Three Phase

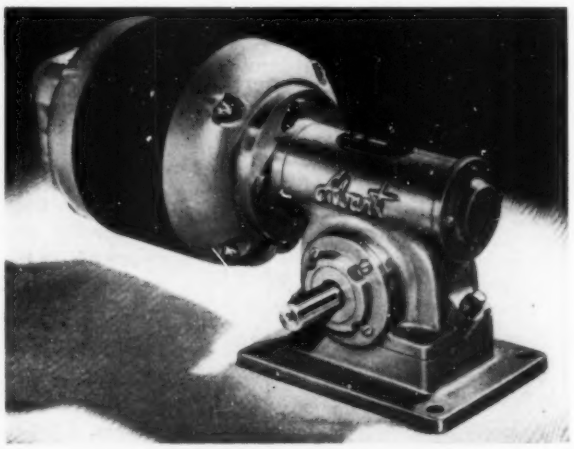
Right angle, single and three phase gear motors are available in $\frac{1}{6}$ to 5 hp at ratios from 5.8:1 to 100:1. Output speeds range from 17.5 to 302 rpm.

Motor mounts on one surface in any position. Nickel bronze gears mesh with hardened and ground worms; the worms are integral with shafts. Neoprene seals protect ball bearings. Alloy steel shafts (180,000 to 320,000 psi, tensile) are ground. Drip-proof, totally enclosed, or fan cooled models are furnished. Brakes are optional. Baked varnish protects windings.

Special voltages and cycles, off-standard speeds, and parallel output or right angle, vertical output motors can be manufactured on special order.

Torque is taken by the semi-steel gear case. Overall lengths vary from 14 to 27 in.; heights from $7\frac{1}{4}$ to $15\frac{5}{8}$ in. Top-mounted capacitors for single phase motors increase height slightly. Drip-proof motors are standard; totally enclosed, non-ventilated or externally fan cooled motors are offered at extra price. Single phase units are 115/230 v, except $\frac{1}{6}$ and $\frac{1}{4}$ hp, which are single voltage. Three phase motors are 220/440 v. 60 cycle is standard, 50 cycle brings extra cost.

Speeds from 302 to 47 rpm are available in $\frac{1}{6}$ to 5 hp, 36 rpm up to 3 hp, 29 and 23 rpm up to 2 hp, and 17.5 rpm at a maximum of 1 hp. Total weights vary from 34 to 235 lb in the 20 standard frames.



Abart Gear and Machine Co., 4828 W. 16th St., Chicago 50, Ill.

Circle No. 6 on Reader Service Card

CONTINUED ON PAGE 170

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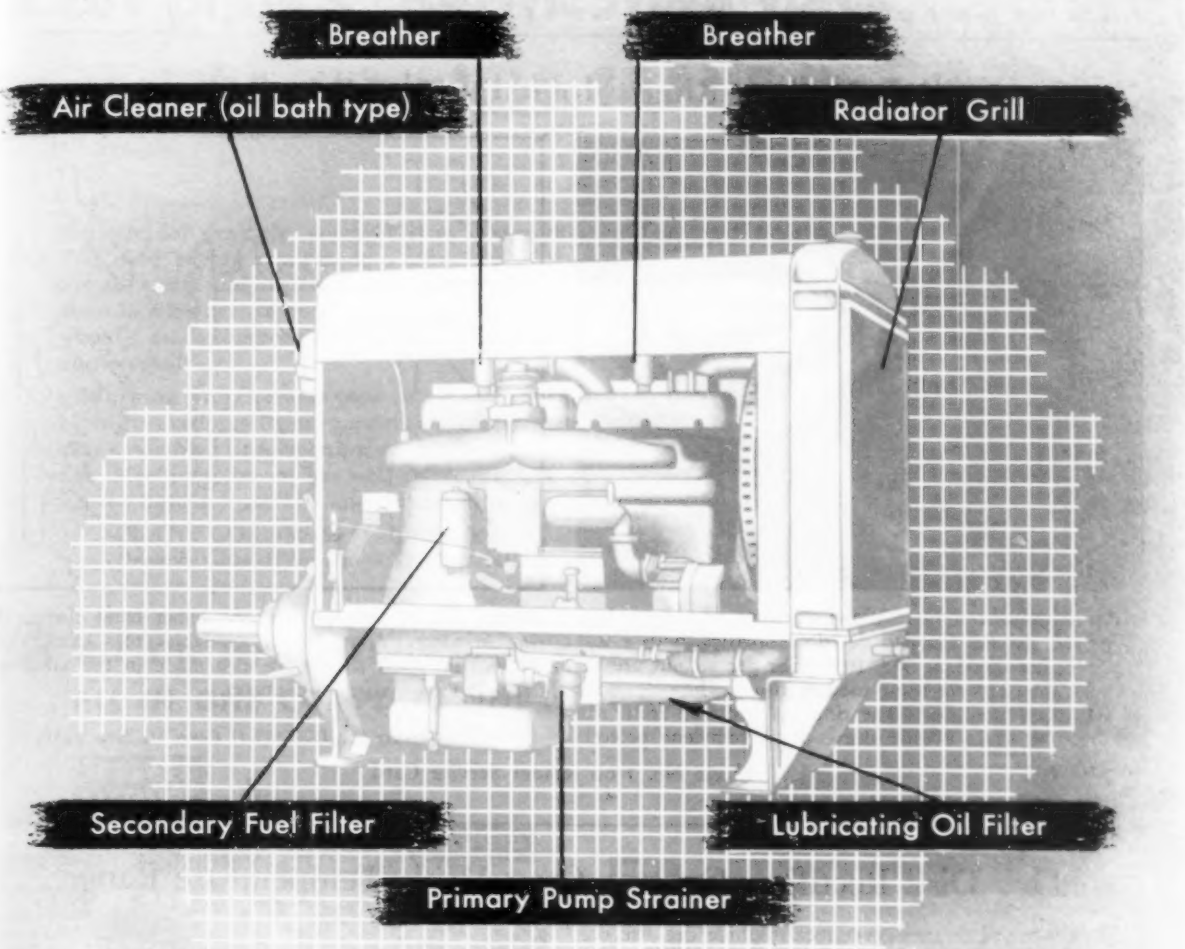


wire cloth

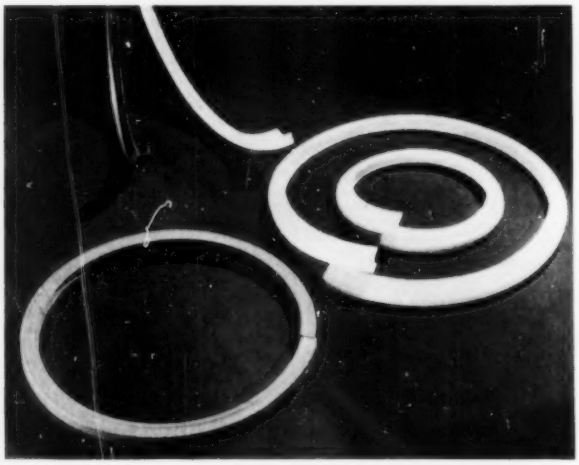
FOR DIESEL POWER UNITS

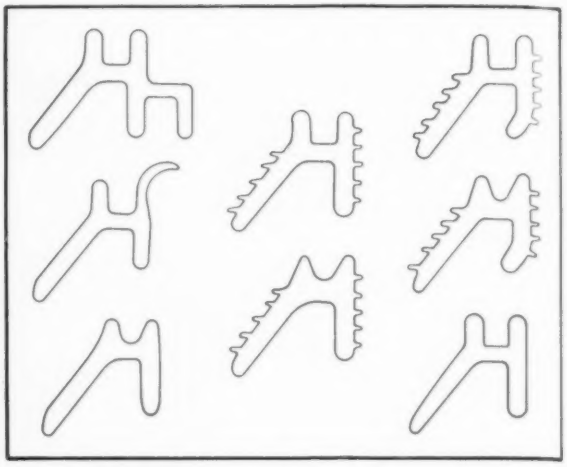
1950 marks fifty-two years of American Diesel development. Four years before Adolphus Busch built the first American Diesel, Horace Reynolds started the first wire cloth factory west of New York state, in Dixon, Ill. Today's modern Diesels would not have been possible without the parallel development and application of component parts and assemblies in which woven wire cloth plays an important role. By promoting engine reliability and economy, Reynolds Wire Cloth is helping Diesel power units to set new records for low power costs. Reynolds more than fifty years of wire weaving experience is at your disposal.

Reynolds Industrial WIRE CLOTH

REYNOLDS WIRE CO.
DIXON, ILLINOIS

Polyethylene Rings Molded To Insulate Tubes





An improved technique allows extrusion of polyethylene in curved shapes. Good dielectric properties, with a loss factor below 0.001 at 3 x 10^9 cycles per second are retained. The rings are being used to insulate the outer edge of the television tubes. The ring fits over the front rim of the tube while a cone sleeve of polyethylene protects the rest of the tube surface. A rubber band clamps the ring in place. A variety of available cross sections are shown. Elliptical shapes are also made in the same sections. 3 ft is maximum diameter.

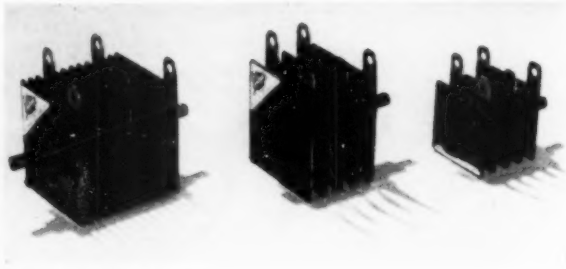
Anchor Plastics Co., Inc., 533-41 Canal St., New York, N. Y.

Circle No. 7 on Reader Service Card

Dry Disk Rectifiers Operate Over Wide Temperature Range

Three sizes of magnesium-copper sulphide dry-disk rectifiers have been introduced. Models include a rectifier with four to two amp capacity, one with six to four amp capacity, and a third for ten to six amperes. These rectifiers have been designed for use in small battery chargers and trickle chargers, and for other low voltage d-c power supply applications.

Temperature operating range is —40 to —284 F. The rectifiers are not affected by atmospheric conditions. Output has been found relatively uniform over operating life.



Electronic Rectifiers, Inc., 2104 Spann Ave., Indianapolis, Ind.

Circle No. 8 on Reader Service Card

Chart Recorder Measures Temperature Or Pressure

Temperatures and pressures under changing conditions are recorded by a 6 in. chart recorder measuring $8\frac{3}{8}$ by $10\frac{5}{8}$ by $3\frac{3}{4}$ in., and weighing $7\frac{1}{2}$ lb. Temperature and pressure systems are interchangeable. The instrument may be converted from a temperature to a pressure recorder, or vice versa, or changed in range of temperatures or pressures recorded.

Mounting brackets are reversible and the recorder may be flush or wall mounted. Legs and handle are also available to make the recorder portable and self-standing. Temperature ranges cover 40 to 1,000 F; pressure system ranges, 0 to 5,000 psi.

The bulb of the instrument is supplied with mercury, gas or vapor filling, according to use. Size of the bulb depends upon the filling and the length of tubing actuated. The instrument is used for measuring temperatures in ordinary rooms and refrigerated rooms, and for checking temperatures in refrigeration installations.

Gotham Instruments, Div. of American Mach. & Metals, 149 Wooster St., New York 12, N. Y.

Circle No. 9 on Reader Service Card

CONTINUED ON PAGE 172

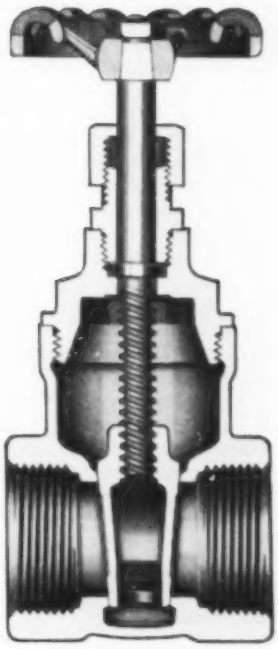
UNIVERSITY OF MICHIGAN LIBRARIES

To cut piping specification time ...use the complete CRANE line

Suppose, for example, your job is to design this *Oil Preheater.* You need piping equipment . . . and lots of it . . . to efficiently control steam and heated fuel oil. Where can you find this equipment in a hurry? In your Crane Catalog, of course. It lists the world's most comprehensive selection of valves, fittings, pipe and accessories.

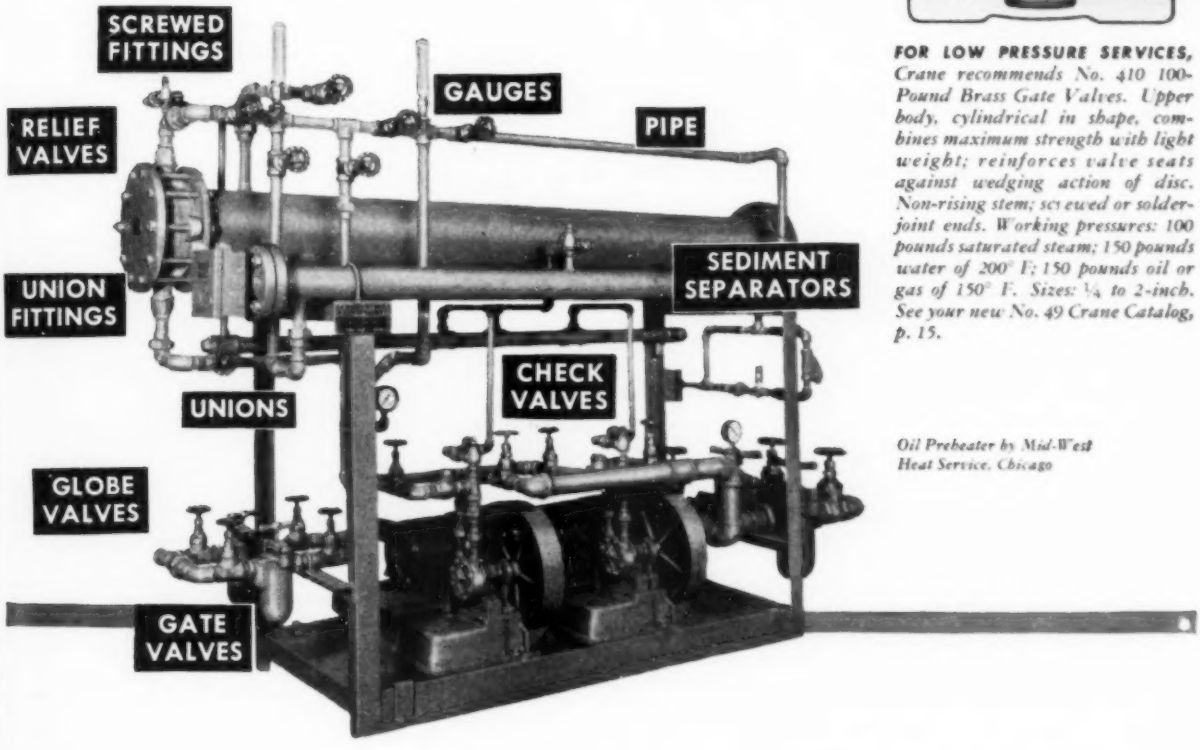
Working with Crane as your single source of supply speeds all piping procedures, from your original design through to final product assembly. Crane equipment gives your design added value, too. Machinery buyers are well aware that Crane Quality stands for the best in materials and construction of piping equipment.

CRANE CO., 836 S. Michigan Ave., Chicago 5, Ill.
Branches and Wholesalers Serving All Industrial Areas



FOR LOW PRESSURE SERVICES, *Crane recommends No. 410 100-Pound Brass Gate Valves. Upper body, cylindrical in shape, combines maximum strength with light weight; reinforces valve seats against wedging action of disc. Non-rising stem; screwed or solder-joint ends. Working pressures: 100 pounds saturated steam; 150 pounds water of 200° F; 150 pounds oil or gas of 150° F. Sizes: ¼ to 2-inch. See your new No. 49 Crane Catalog, p. 15.*

ALL THE PIPING EQUIPMENT YOU NEED ... ALL IN ONE COMPLETE LINE ... CRANE



Oil Preheater by Mid-West Heat Service, Chicago

EVERYTHING FOR EVERY PIPING SYSTEM **CRANE**

VALVES • FITTINGS • PIPE • PLUMBING AND HEATING

Servo Amplifier Completes Model Loop for Laboratory

Servo designers have been offered a servo amplifier to be combined with the manufacturer's servo motor to give a complete model of an actual servo loop. This servo circuit is set up in a fraction of the time needed to build the actual circuit. Only physical dimensions are standardized. The engineer specifies such characteristics as phase voltage, reduction rate, mounting, gain, phase shift, input impedance and damping to fit the particular servo loop under study.

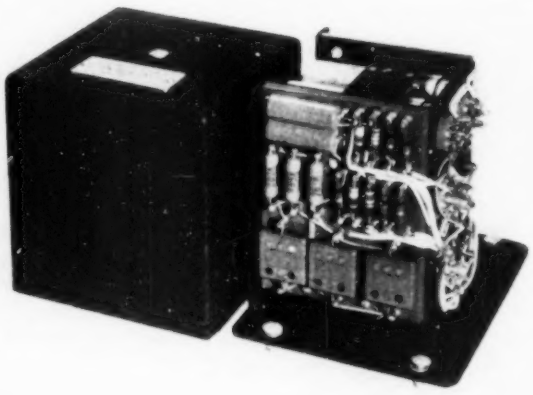
The amplifier drives a servo motor wound for plate to plate operation. Two types of the complete package are available.

1. A laboratory model proves the circuit performance of a loop under study. It can not meet environmental specifications. The engineer completes circuit development on one economical setup.

2. A plug-in package consists of the same circuits used in laboratory model. The design meets military and industrial environmental specifications. Applications are in optical positioners, remote indicators, fire control computers, velocity servos, process controllers and military trainers.

Identical circuits are used in the laboratory development amplifier and in the complete operating amplifiers. Damping is usually designed into amplifier circuits.

In carrier frequency servo loops, the amplifiers give all the circuitry needed in the error signal path. They operate with autosyns and both resistive and inductive potentiometers. System response shows a high damped frequency, low errors and some insensitivity to carrier frequency shifts with light inertia loads. With d-c input signals, a d-c to a-c modulator is required ahead of amplifier in error path. Operates on 60 or 400 cycles.



Transicoil Corp., 114 Worth St., New York 13, N. Y.

Circle No. 10 on Reader Service Card

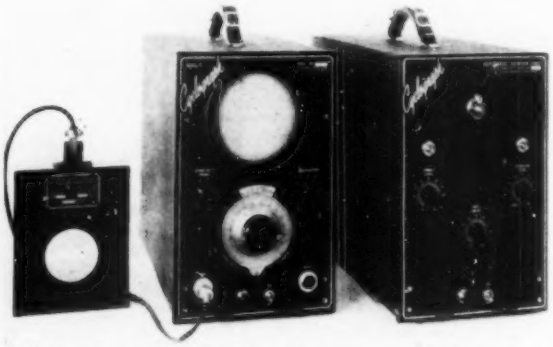
Metal Parts Are Sorted By Metallurgical Characteristics

High speed non-destructive quality control and sorting of metals parts is accomplished by a new automatic instrument set. Operating on magnetic principles, it can sort by analysis of part structure, hardness, case depth, brittleness or almost any metallurgical difference. Speeds of 1 to 5 parts per sec are attained on belt or other fast feeders. Parts reading below a given value can be rejected, or the instrument can reject all parts not within a set range.

The oscillograph utilizes the principle that a metal core introduced into a high frequency field of a test coil will cause energy losses. It is fundamentally an oscillator, or multivibrator. A sensitivity control governs the amount of regenerative feedback and allows adjustment of the initial energy output to any desired level.

The test coil, which is part of the tuned circuit, is bifilar wound to obtain maximum mutual inductance between the sections of the oscillator and to provide self-shielding from outside interference. This coil controls the operating frequency of the oscillator. Core losses occuring in the field of the test coil decrease the output of the oscillator. This output is viewed on a cathode-ray tube screen.

The pattern appearing on the screen can be correlated with the metallurgical characteristics of the test piece. A reading is taken on a predetermined standard piece having known and desirable characteristics. Metallurgical differences between the predetermined standard and other pieces inserted in the test coil will immediately indicate by a variation in height of the pattern on the screen.



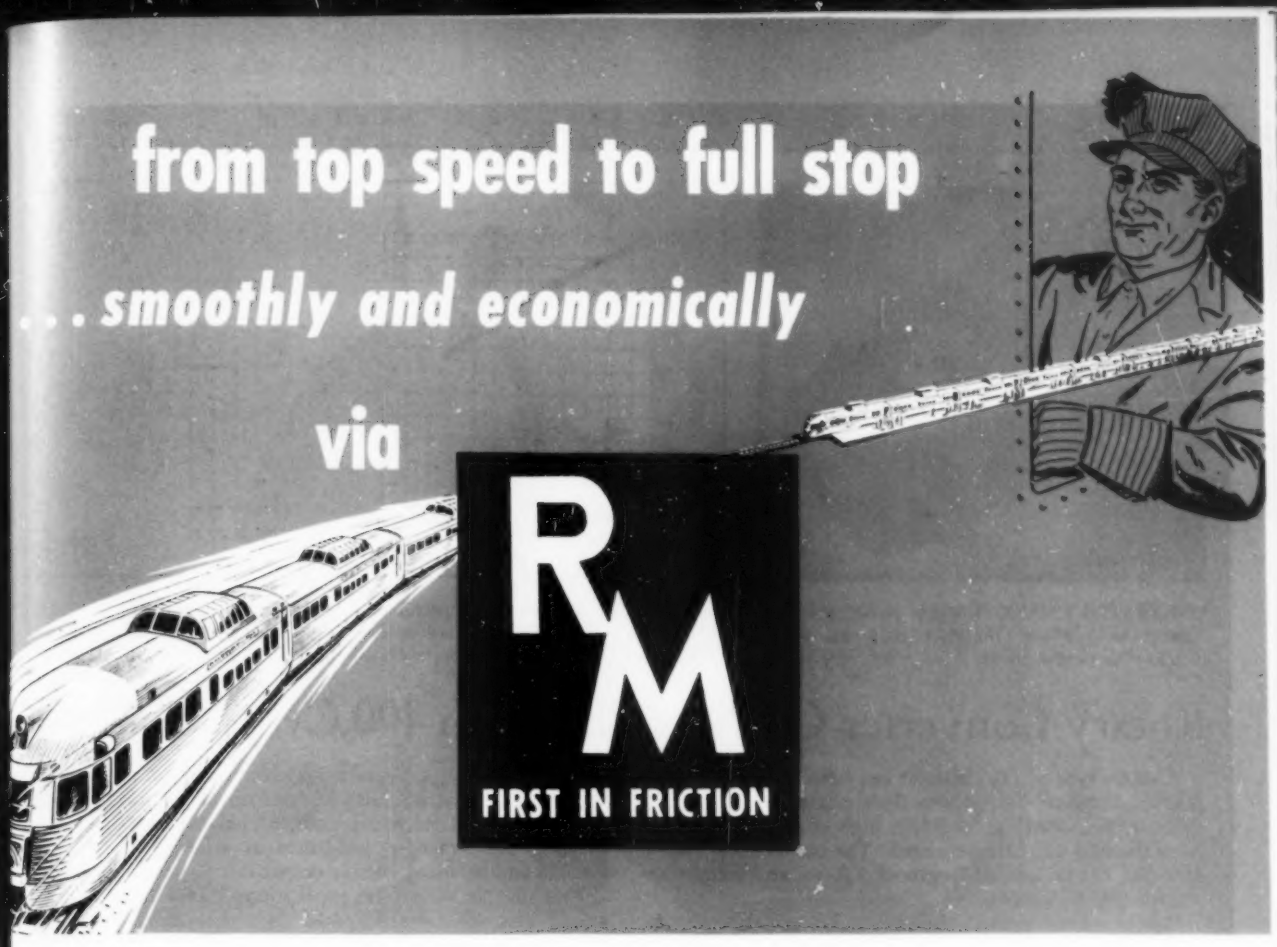
The automatic sorter automatically takes the "intelligence" from the oscillograph screen. The signal operates relays which either do or do not deliver a 115 v, 60 cycle signal to external sorting gates.

J. W. Dice Co., 1 Engle St., Englewood, N. J.

Circle No. 11 on Reader Service Card

CONTINUED ON PAGE 174

UNIVERSITY OF MICHIGAN LIBRARIES



The California Zephyr and several other leading trains throughout the nation are now operating with an advanced type of brake system.

It is the Budd Disc Brake . . . the first basic improvement in railroad braking since the invention of the air brake. How fast does it operate? Here's an example. *With disc brakes, in which R/M friction materials play a vital part, an 80-ton car can be brought from 100 m.p.h. to full stop in less than half a mile!*

All this is done quietly and smoothly, without squeals or screeches, without jerks or jars. And with vast savings in maintenance costs!

Time and again, in railroad as well as in automotive and industrial applications, R/M has helped pioneer far-reaching improvements in brake and clutch operation. The experience thus gained, plus the facilities of the world's largest producer of friction materials, is at your service when you call in your R/M representative.

RAYBESTOS-MANHATTAN, INC.

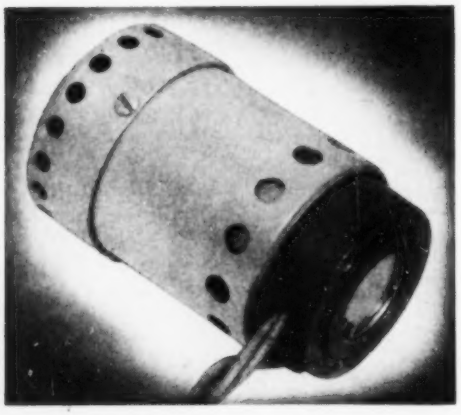
EQUIPMENT SALES DIVISION **620 Fisher Bldg., Detroit 2, Mich.**

445 Lake Shore Drive, Chicago 11, Ill. 4651 Pacific Blvd., Los Angeles 11, Calif.

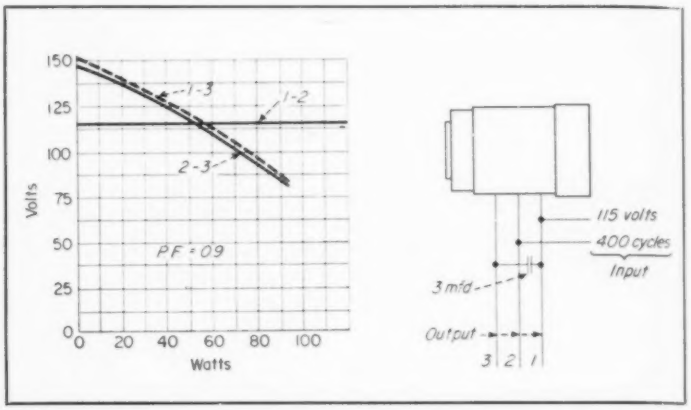
1071 Union Commerce Bldg., Cleveland 14, Ohio

Factories: Bridgeport, Conn. Manheim, Pa. Passaic, N.J. No. Charleston, S.C.

RAYBESTOS-MANHATTAN, INC., Manufacturers of Brake Linings • Brake Blocks • Clutch Facings Fan Belts • Radiator Hose • Mechanical Rubber Products • Rubber Covered Equipment • Packings Asbestos Textiles • Powdered Metal Products • Abrasive and Diamond Wheels • Bowling Balls



CONVERTER CHANGES single phase, 400 cycle current to 3 phase. Diameter is 2 in., length 3 3/32 in., weight 16 oz.



WITH WIRING SHOWN, power factors from 0.6 to 1.0 are carried. Bleeder resistances are connected to ensure balanced output where load is smaller than specified. There are no brushes or commutators to be serviced.

Rotary Converter Changes Phases on 400 Cycle Current

A rotary type phase converter has been designed to convert single phase 400 cycle to three phase 400 cycle. Balanced output, stability, long life, high efficiency and minimum size and weight are claimed. The converter is 2 in. in diameter, has an overall length of $3\frac{3}{32}$ in. and weighs approximately 16 ounces.

Output voltage in this converter, as in any similar device, is a function of the load and power factor. However, for power factors ranging from 100 to 60 percent, a balanced three phase output can be obtained for a given load condition. As high as 80 va with unbalance not exceeding 10 percent can be delivered.

In cases where the actual load is less than that necessary for balanced output conditions, bleeder resistances connected in delta or wye may be used on the output side of the converter to reach the point of balanced output. By utilizing these bleeder resistances as voltage dividers, a variable and balanced voltage output can be obtained.

This unit is suited for applications where high hp to weight and volume ratios is desired. It can be used in gyroscope systems, rectifier units, frequency control circuits, control systems and electronic equipment. The lack of commutation and brushes keeps servicing to a minimum and removes all possibilities of RF interference.

Eastern Air Devices, Inc., 585 Dean St., Brooklyn 17, N. Y.

Circle No. 12 on Reader Service Card

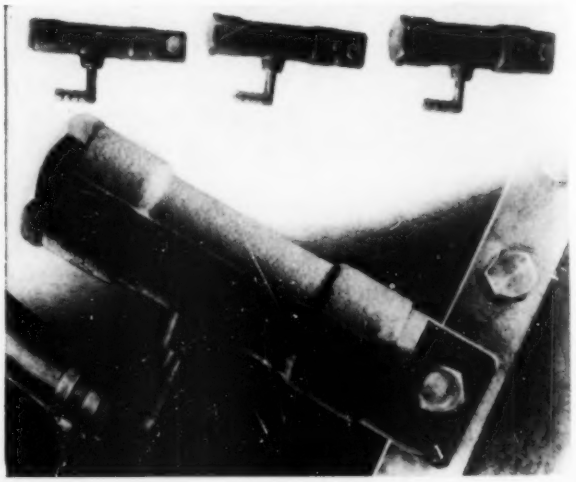
Bronze Liner Lubricates Vibrator Cylinder

The availability of pneumatically-operated vibrators designed for industrial applications, has been announced. Available from stock in eight sizes, vibrators are recommended for removing patterns from sand molds, providing steady flow of powdered or granular materials, feeding small parts, settling the "mix" in concrete block and vault machines, vibrating packaging tables, moving granular materials through tubes and chutes, and fatigue testing.

A special bronze alloy cylinder liner is claimed to assure positive and permanent piston lubrication. On 80 psi air line pressure, 2,200 to 6,000 impacts per minute, depending upon piston diameter, are delivered. Speed of vibration can be regulated by line pressure. Air consumption ranges from $4\frac{1}{2}$ to 8 cubic ft per minute. In addition, pistons ends are uniformly hardened. Piston diameters of eight sizes range from $\frac{1}{2}$ to $2\frac{1}{2}$ inches.

SPO Inc., 6554 Grand Division Ave., Cleveland 5, Ohio

Circle No. 13 on Reader Service Card



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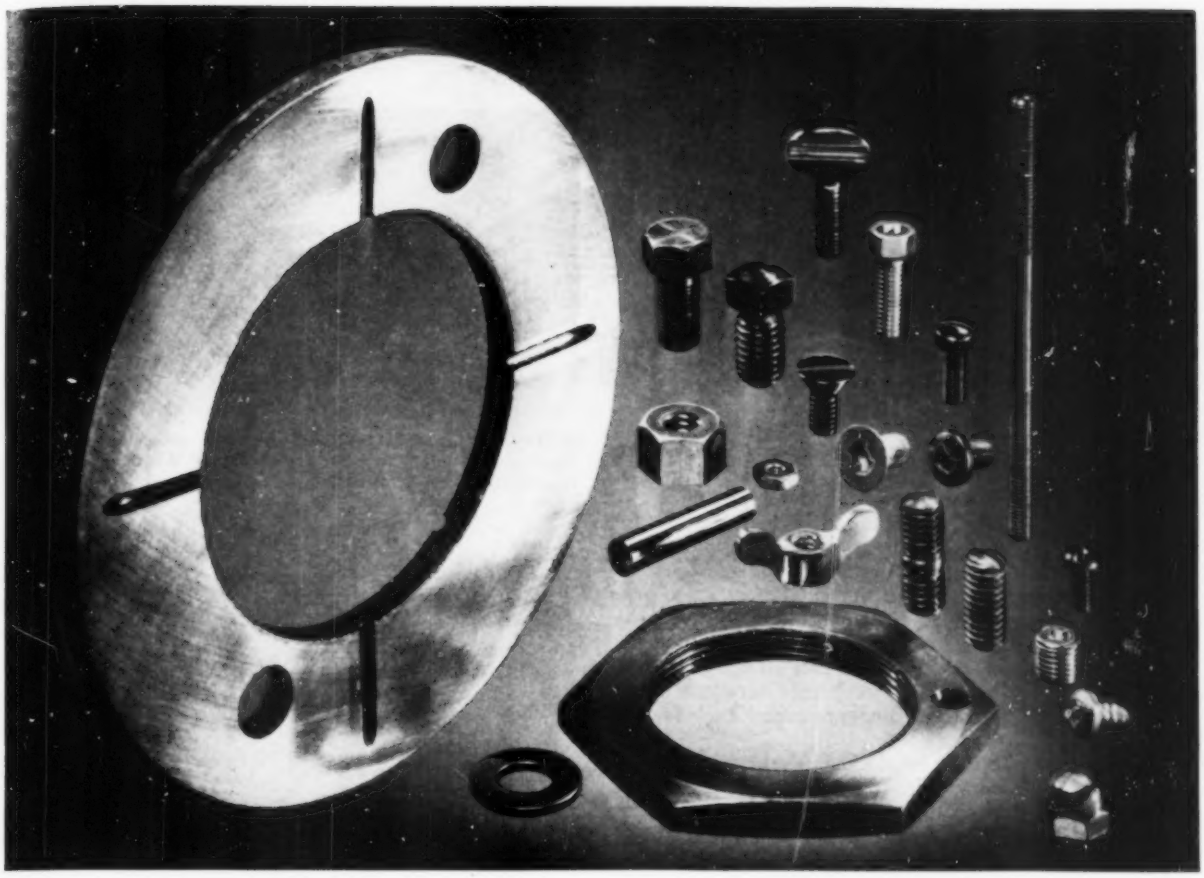
UNIVERSITY OF MICHIGAN LIBRARIES



STAMPINGS

From Our Pressed Metal Products Plant

CLUTCH HEAD SCREWS

In addition to CLUTCH HEADS and a complete family of cold headed fasteners. **WE SPECIALIZE IN WASHERS, SHIMS AND BLANKS** from $\frac{1}{8}$" in diameter to 13" inclusive . . . produced from .002" to $1\frac{1}{4}$" thick material.

Thus, with long experience and adequate equipment, **HOLES OF NEARLY EVERY CONCEIVABLE SIZE AND SHAPE** can be held to either commercial or very exacting limits from **STOCK TOOLS.**

Among services not usually offered by stamping plants is United's extended use of secondary operations . . . combining up-to-the-minute heat treating and machining operations with volume production facilities for grinding, tapping and related processing refinements.

For important cost advantages, consider the re-design of your present screw machine, forged, cast or molded parts to the use of flat, formed or extruded stampings.

WE AIM TO CARRY THE MID-WEST'S LARGEST STOCK OF SHEET, STRIP AND PLATE IN LOW AND HIGH CARBON STEEL . . . ALSO IN ALLOYS

UNITED SCREW AND BOLT CORPORATION

CLEVELAND 2 — CHICAGO 8 — NEW YORK 7

Line of Safety Switches Covers Range of Industrial Uses

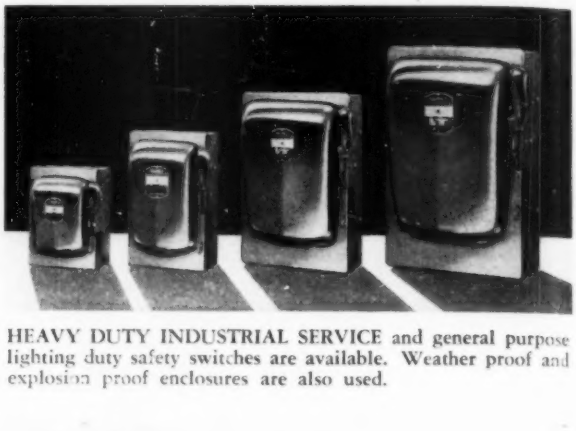
Padlocks hold switch and cover in position. Hot terminals are fully covered.

Full production has been announced on a line of safety switches, with three types covering heavy-duty industrial, standard industrial, and general purpose applications.

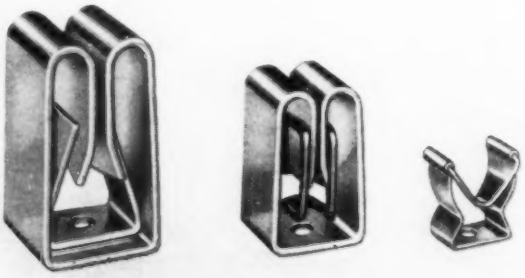
The heavy industrial switch capacities range up to 2,400 amps at 250 v and 1,200 amps at 600 v, in fusible or non-fusible models. Concealed blade types are offered in addition to the visible blade switch shown. Plunger action gives compactness, with the advantages of knife blade action. Current carrying parts are copper, completely silver plated. Wax impregnated hemite, a mixture of asbestos and Portland cement is used in the arc barrier insulator of the switch which swings out on hinges to expose back for easy wiring.

The switches are fully interlocked. The interlock can be neutralized with a screw driver for inspection under load. The interlock engages the operating handle and can be removed easily. If the cover is closed with the switch in the "on" position, the interlock automatically engages in the next operation.

A complementary line of concealed blade switches is offered. Construction features are comparable. The entire base is removable; action is double break. High rupturing capacity is claimed. All copper parts are cadmium-plated.

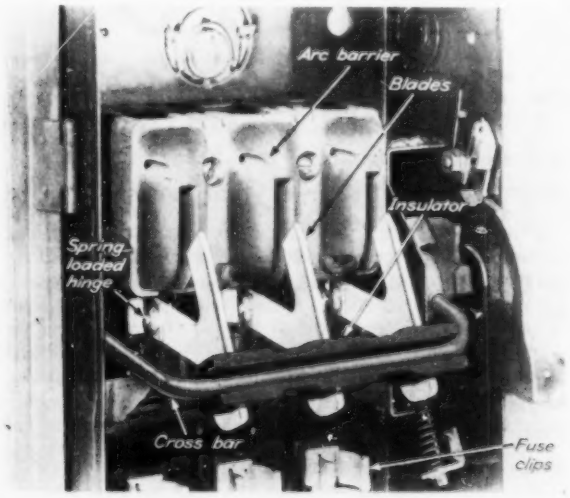


HEAVY DUTY INDUSTRIAL SERVICE and general purpose lighting duty safety switches are available. Weather proof and explosion proof enclosures are also used.

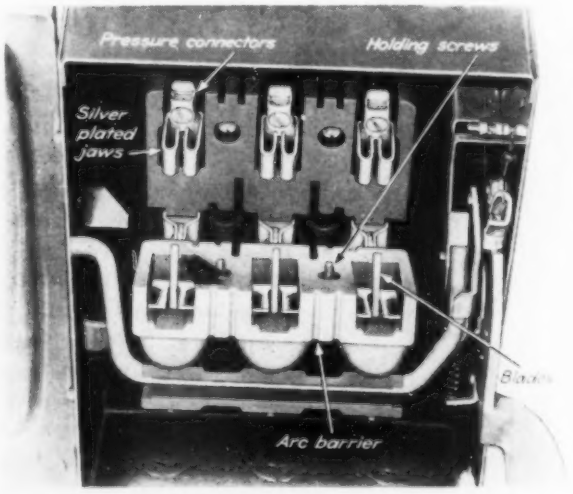


POSITIVE PRESSURE FUSE CLIPS, furnished in all switches, reduce heating at fuse contacts. Copper contacts increase conductivity and steel springs give contact pressure.

Square D Co., 6060 Rivard St., Detroit 11, Mich.



CROSS BAR is steel, protected from the blades by melamine insulator. Spring loaded blades, visible in "off" position plunge into arc chamber through small openings. Hot line terminals and blade jaws are completely covered in "On" position. With the switch "Off" all exposed parts are dead and can be visually checked as such by visable blades.



MAGNETIC PLATES (arrow) blow out arc. With switch open, backing off holding screws allows arc barrier to swing back. Insulation used in the switch and fuse base is wax impregnated molded hemite. Pressure wire connectors are removable from the front. Gutter space is free of obstructions and sharp corners which might injure wire insulation.

Circle No. 14 on Reader Service Card

CONTINUED ON PAGE 178

UNIVERSITY OF MICHIGAN LIBRARIES

Mr. Parks had a tubing problem . . .

so he called in the ELECTRUNITE Metallurgist . . .

and in practically no time at all . . .

Mr. Parks manufactures school desks. Very good desks, too . . . but, even so, Mr. Parks is always looking for ways to make them better.

This is the way Mr. Parks' problem came about . . . 1½" x 11 gauge tubing is bent U shape to form a support for both seat and table of Mr. Parks' desks. Design calls for a short radius bend, and he was using a fully normalized tube that Rockwell "B" tested under 55.

"Not stiff enough", decided Mr. Parks. "No desk of mine is going to sag when a heavyweight leans against it! On the other hand, the tube must not be so hard it breaks in bending." What to do? — one thing, of course, and Mr. Parks did it. He called in the ELECTRUNITE Tubing Metallurgist. As a result of this call, processing changes were made resulting in a stiffer tube with a higher yield point and a higher Rockwell reading. These tubes readily take the short radius bend and still give Mr. Parks' desks the extra sturdiness and strength he wanted them to have.

Mr. Parks isn't his real name, of course, but his problem was real. When you have a tubing problem, why not do as Mr. Parks did? Call for a Republic ELECTRUNITE Tubing Metallurgist. There is no charge for his services and no obligation on your part for his help. Just let us know when you would like to see him—and he'll be there.

REPUBLIC STEEL CORPORATION
STEEL AND TUBES DIVISION • CLEVELAND 8, OHIO
Export Department: Chrysler Building, New York 17, New York

Republic
REPUBLIC STEEL
ELECTRUNITE TUBING

Polyester Plastic Coil Block Increases Solenoid Life

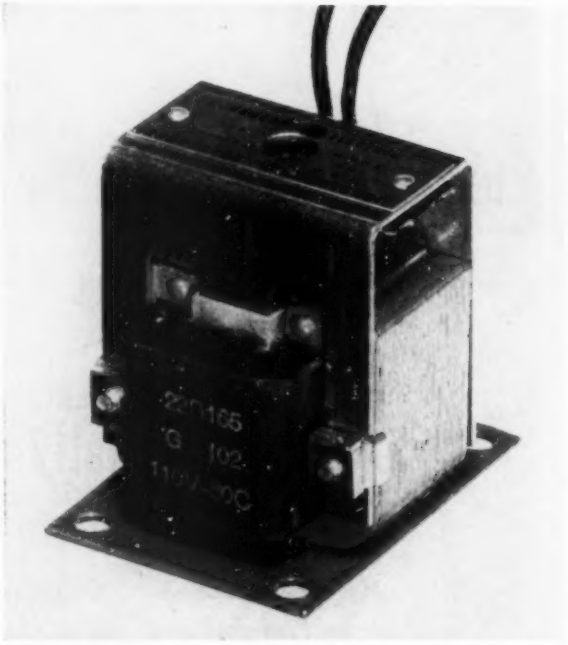
A short-stroke solenoid which tests indicate to have a life of 5 to 10 million operations, is now being marketed. It features a cast-permafil coil assembly that is the result of years of research and development, according to the manufacturer.

The basic component is a paper layer-wound coil that holds the voltage between turns to a minimum and reduces the possibility of failure due to reduced voltage between layers of turns. Two 12 in. flame-resistant leads are firmly anchored and taped, and the coil is placed in a mold into which a liquid polyester resin is poured. Reactors are then added, and in about two minutes the resin sets. Finally, the coil assembly is baked. This results in a solid block around the coil that won't crack and is moisture-, oil-, shock-, and vibration-proof.

Originally designed for high-speed, pilot-valve operation, the solenoid is suitable for any heavy-duty industrial application requiring a push-type solenoid with a maximum stroke below $\frac{5}{16}$ in. It is available in four models, including strokes of $\frac{1}{8}$ in. and $\frac{5}{16}$ in., and two stacking thicknesses.

The device has a "T" magnet, and a large dual-pole shader keeps operation quiet. All surfaces are precision machined. Other features include a heavy stainless-steel backstop assembly with an oil- and water-resistant neoprene pad fastened to it to absorb the shock of the return stroke. The coil can be changed without difficulty.

Comparative life tests indicate that the solenoid will operate from 5 to 10 million times before failure. Three other solenoids used in the tests for comparison failed, either partially or completely, between 500,000 and 800,000 operations.



SHORT-STROKE SOLENOID HAS a test-indicated life of 5 to 10 million operations. A cast polyester resin gives a moisture-, oil-, shock-, and vibration-proof coil block which is the secret of the increased life span.

General Electric Co., Schenectady 5, N. Y.

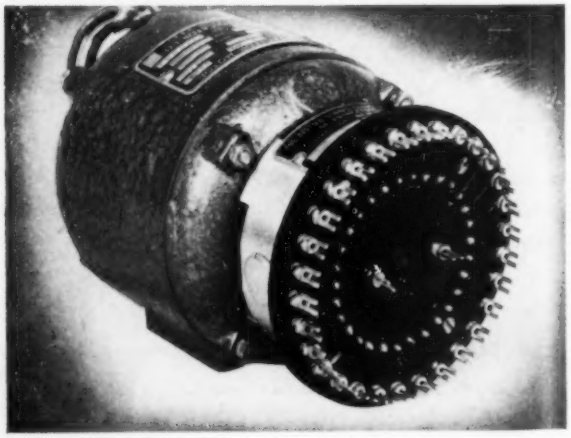
Circle No. 15 on Reader Service Card

Rotary Switch Samples 30 Channels in Synchronism

This multichannel sampling switch, composed of a motor and a switch, has been designed in a single package, weighing 53 ounces. Overall dimensions are 5 in. long by $3\frac{1}{2}$ in. wide by $3\frac{9}{16}$ in. high.

The synchronous driving motor is rated at 1,800 rpm, 115 v, single phase, 60 cycle a-c. Since the switch is driven directly by the motor shaft at a rate of 30 revolutions per second, sampling of all 30 "break-before-make" channels is in synchronism with the a-c line viltage. This synchronous switching is suited for the display of the sampled information on a CRO. The 30 active contact pins are connected to terminal lugs to facilitate wiring by the user. A similar unit is available with all 60 contacts made accessible. With its shorting type wiper, this device features versatility in possible switching arrangements.

The 30 contact switch connects each of the 30 terminals singly, and in order, to a single sampling contact at the center. The 60 contact switch has no center terminal and each terminal has a mate. Mated terminals are connected in succession.



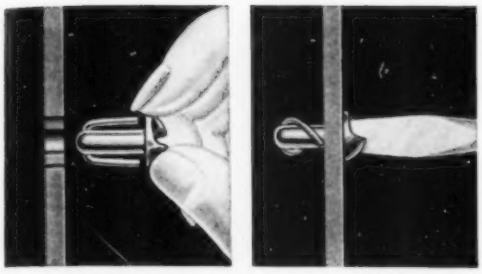
Applied Science Corp. of Princeton, Box 44, Princeton, N. J.

Circle No. 16 on Reader Service Card

CONTINUED ON PAGE 180

UNIVERSITY OF MICHIGAN LIBRARIES

Have you a similar use for this 1-piece fastener?

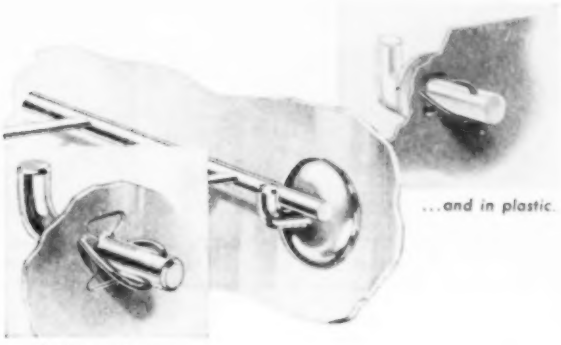


It's a Blind Rivet

...Or a removable fastener. It locks and unlocks with a 90° clockwise rotation. No mating parts such as nuts or receptacles are required.

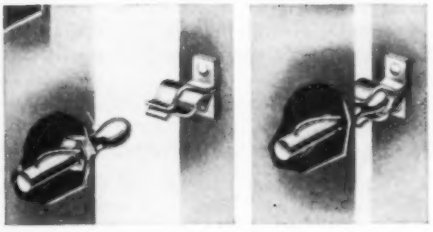
It's a Shelf Support

...For ranges or refrigerators—in plastic and metal. Leading appliance makers have achieved substantial installation savings through its use.



...and in plastic.

Both in metal...

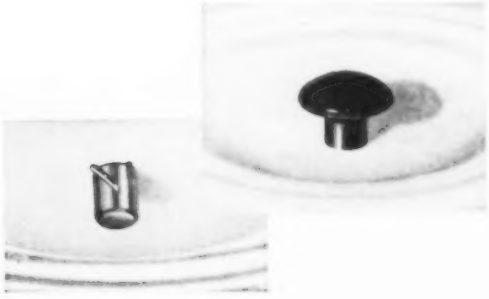


It's a Cabinet Door Strike

...Simple to install; eliminates welding and cuts assembly cost. Any head can be designed without affecting fastening principle.

It's a Lifter Knob or Dashboard Plug

...Plastic Spring-Lock heads are molded around steel inserts, giving strength at point of load or impact. Any shape head can be molded in any color.



What's Your Application?

...Tell us how you can use Spring-Lock Fasteners in your products. We'll be glad to work out the details with you.

SIMMONS FASTENER CORPORATION

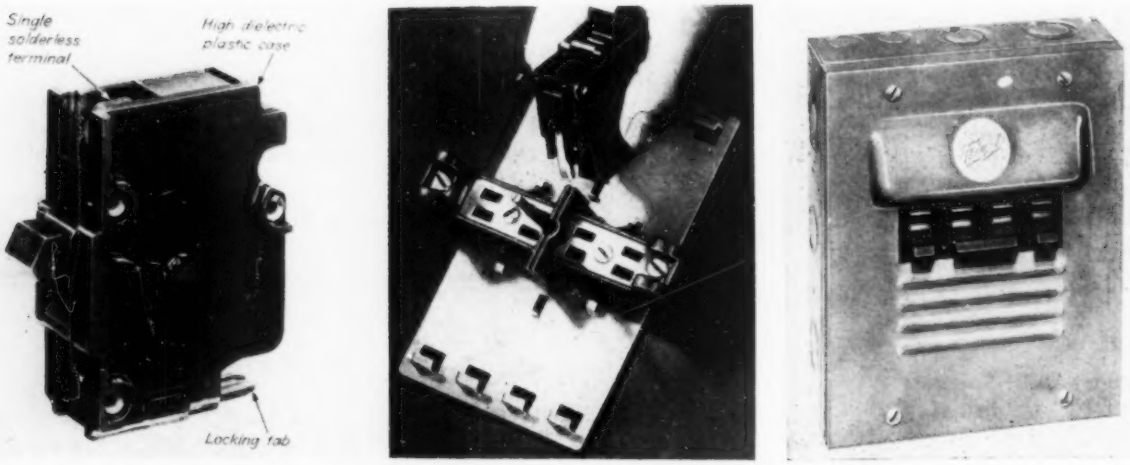
1751 NORTH BROADWAY, ALBANY 1, N. Y.

Simmons Fasteners

QUICK-LOCK...SPRING-LOCK...ROTO-LOCK

Send for more data and Free Samples today.

Standard Breakers Snap Into Enclosure Base



SIMPLE CIRCUIT BREAKER UNITS are designed with side plates, two springs, two assembled parts, and a handle. Units lock in enclosure without screws. Double pole unit shown in box at right throws two breakers simultaneously, manually or by internal bar link. Nine enclosures are standard with up to 20 circuits, carrying 30 to 100 amps, 2 and 3 wire.

Mass produced circuit breaker units give a wide variety of combinations with only nine enclosures. Standard units snap into place without screws, and there is only one terminal to connect. An insulated neutral is carried in each enclosure.

Breakers work on combination thermal-magnetic trip action. In the magnetic action, an electro-magnet breaks the circuit when current rating is greatly exceeded. The thermal action uses a bimetal which flexes and trips the breaker when a predetermined current-time rating is exceeded. Harmless momentary current surges—as in motor starts—are passed, but the circuit is broken on shorts or dangerous overloads.

Double pole circuit breakers are also offered. These are tied together, not only by a single handle, but also by an internal link bar. One circuit cannot be broken without breaking the other.

Enclosures for up to 20 circuits, carrying from 30 to 100 amps, 2 and 3 wire, are available. These are designed to replace a wide range of the manufacturer's controls now in use.

Federal Elec. Products Co., 50 Paris St., Newark 5, N. J.

Circle No. 17 on Reader Service Card

Air Cylinder Gives Long Stroke With High Power

An air cylinder has been designed and developed to meet demand for a positioning device with a long stroke and high power. Applications in damper positioning on diesel engines, controlling rheostats by means of a rack and pinion, and damper positioning on pneumatic temperature and pressure control systems are expected.

The cylinder operates with a supply air pressure of 80 psi. Supply air pressures as great as 150 psi may be used but cylinder rating is not increased. However, the use of an external spring, together with higher air pressures, up to 150 psi, increases the clinching force in the end positions.

Control air pressure force acting on the diaphragm is opposed by the force of the spring, determined by the position of the motor piston rod. When these two forces are not in balance, high pressure air is either admitted to, or exhausted from the piston cylinder through the feed and exhaust valves.

The motor piston is spring loaded and assumes a position determined by the forces exerted by the air input on the one side and the spring on the other. As the piston moves, a positive positioning linkage attached to the piston rod either compresses or extends the spring tending to rebalance the diaphragm. When the diaphragm is balanced, air pressure to the cylinder is shut off and the piston is stopped.

Holding force is 68 lb at either end position, 230 lb at mid-position of piston. Starting force is 370 lb at either end, 230 at mid-position. Linear stroke is 4¼ in. non-adjustable. Control action utilizes control air input and high pressure supply air to provide positioning output air for the cylinder.

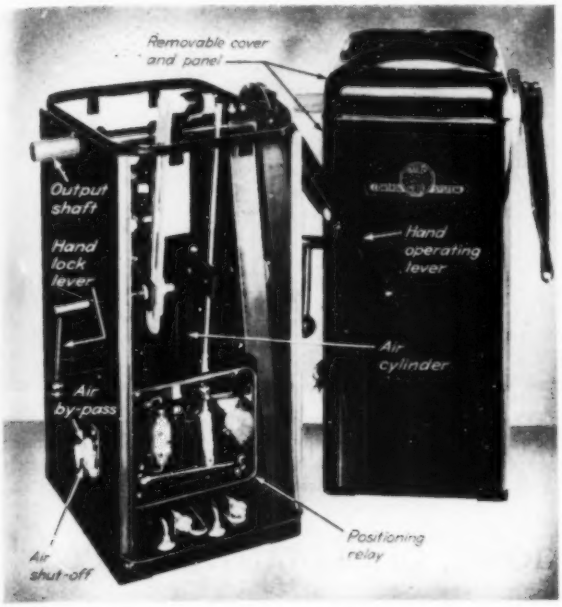
The starting point of the cylinder is adjusted by a screwdriver for from 3 to 13 psi control pressure. The range through which control air varies for a full piston stroke is adjustable from 5 to 19 psi. Normal control air range is 0 to 15 psi, maximum is 30 psi.

Brown Instruments Div., Minneapolis-Honeywell Regulator Co., Philadelphia 44, Pa.

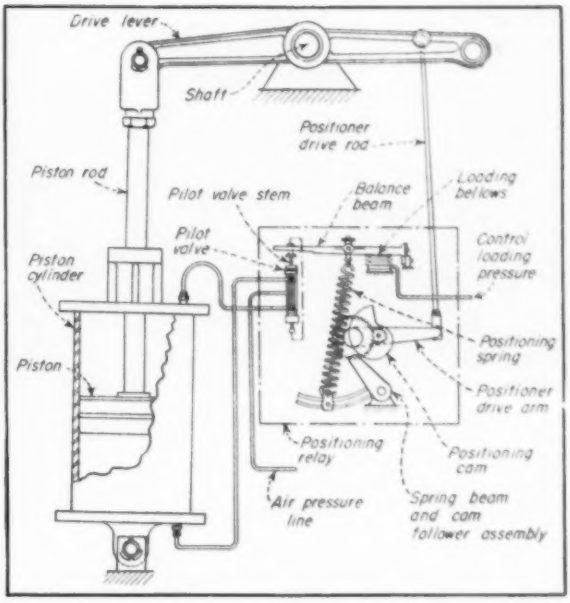
Circle No. 18 on Reader Service Card

CONTINUED ON PAGE 181

UNIVERSITY OF MICHIGAN LIBRARIES



OUTPUT SHAFT POSITION depends on control pressure. Change of 0.10 psi gives maximum actuating torque. Shut-off and by-pass valves and hand lever allow manual operation. Cylinder is double acting, rotating output shaft through a lever. Position locks, signals and standard linkage are offered.



LINE PRESSURE FEEDS to cylinder through pilot valve. Control pressure and the positioning spring oppose each other to move valve. Each control pressure has equivalent shaft position. Position of shaft can vary as first power, square or square root of control pressure, depending on shape of positioning cam.

Pneumatic Relay Controls Air-Operated Positioner

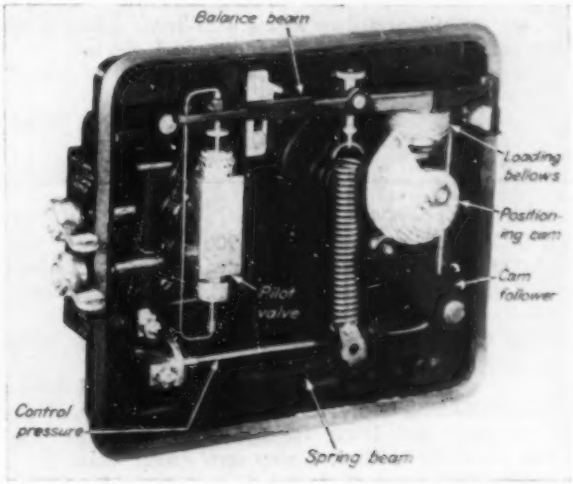
Tiny control pressure gives high torque. Final output shaft position is function of control pressure, following first power, square or square root of pressure.

This air-operated drive applies line pressure to change the position of a shaft on a small change in control pressure. Essentially it works like an electric relay, changing a large output by means of a small control circuit. It consists of a double acting cylinder with rigid supporting stand and enclosure, a positioning relay, and a manual operating wheel or lever. Shaft rotation is 90 deg.

The positioning relay incorporates a cam whose shape determines the relation of drive shaft position to control loading pressure. Three cams are provided to give output positions proportional to the control pressure directly, to its square, or its square root. Modifications are possible.

The shaft can be locked in an extreme position by an adjustable stop, released by a manual air push button. Signal lights can also be arranged. Direction of drive travel can be reversed or drive can be adjusted to remain stationary for any change in control pressure up to 15 psi, or can be set for full travel at the same pressure. One-half travel for control pressures of 5 to 25 psi can be had with simple adjustments. Integral switches can be furnished to make contact at pre-selected drive positions.

An assortment of standard connecting linkage may be ordered directly, to avoid the necessity of detailed drive layouts. Position signals are also extra equipment.

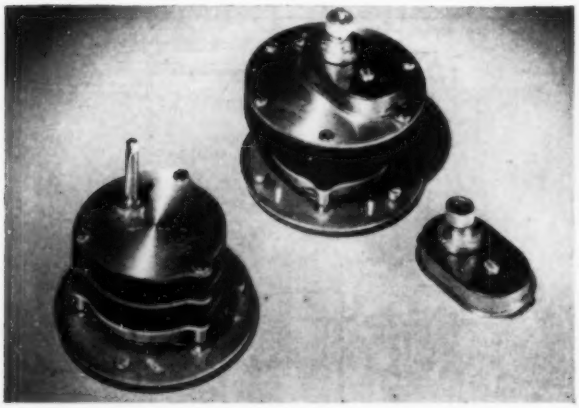


RELAY SUPPLIES NO AIR PRESSURE to cylinder with pilot valve in neutral. Valve stem moves with balance beam, controlled by resultant of bellows and positioning spring forces.

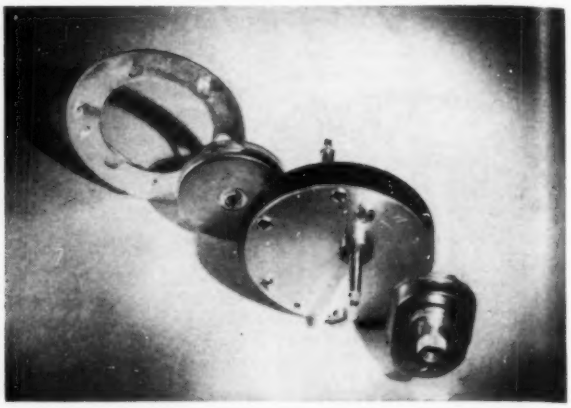
Bailey Meter Co., 1050 Ivanhoe Rd., Cleveland 10, Ohio

Circle No. 19 on Reader Service Card

CONTINUED ON PAGE 182



AT LEFT is 24-hour-wind midget drive. The 8-day-wind drive is shown in the center with gearhead or "turret" mounted to give rotational speed other than 24 hour basic. Detached turret is shown at right. Enclosed escapement is behind gear case.



EXPLODED VIEW of 8-day-wind mechanical instrument drive shows turret. Both mechanisms measure 2 21/32 from mounting ring (in back) to shaft tip. Outside diameter is approximately 3¼ in. in each case. Direction of shaft rotation is optional.

Instrument Drives Give Choice of Rotating Speeds

Two mechanical instrument drives for meters, and other apparatus are characterized by escapement mechanisms which can be detached for service or replacement. Overall diameters are 3¼ in. and 3 3/16 in. respectively; both models require 1 3/32 in. between the plane of the driven chart (the rotation of circular dials and recorder charts being the most common use for such mechanisms) and the mounting ring.

The larger model has an 8-day wind and a basic 24-hour rotation. Gear heads snap on the main arbor to convert to 2 hour, 3 hour, 4 hour, 6 hour, 8 hour, 12 hour, 48 hour, 3 day, 7 day and 8 day rotation. The smaller model has a 24 hour wind with a torque which drops from 8.6 to 4.7 in-lb between full-wind and 24 hour rundown.

Capacity is provided to give 1.8 in-lb of torq
hour rundown. Both models wind through the hub
The detachable escapement is sealed in clear

This subassembly twists and snaps into place without tools, controlling main spring action through a simple connecting square. Escapements may be bought separately to control other devices. The standard mechanism permits one turn of the connecting square per hour, while rates of 2 min and 4 min are available on special order.

Both models mount by snap .action on a circular plate of 3¼ in. dia, which in turn may be mounted on bosses in the instrument case. Main arbors fit standard chart hubs, and are directly tied to the mainspring. Drive cases are die-cast aluminum and sealed; escapement cases are stainless steel and clear plastic. Gears are mounted in phosphor-bronze bearings. Direction of rotation is optional, counter clockwise being furnished unless otherwise specified. Drive self-starting and can be permanently mounted.

Pittsburgh Equitable Meter Div., Rockwell Mfg. Co., Pittsburgh 8, Pa.

Circle No. 20 on Reader Service Card

CAPACITORS are designed for 900 to 1,000 watt photo-flash lamps, rated at 500 v, to give 100 watt-sec energies. Capacity is 200 mfd. Cornell-Dubilier Electric Corp., South Plainfield, N. J.

Circle No. 21 on Reader Service Card

SOLENOID PNEUMATIC VALVES are offered 3 and 4 way for 200 to 1,500 psi at 18 to 28 v, 1 amp continuous. Tube sizes ¼, ⅜ and ½ in. Rhodes Lewis Co., 3652 Eastham Dr., Culver City, Calif.

Circle No. 22 on Reader Service Card

STRAIN GAGES are self-compensating for temperature. Single element is stock; rosette and double element also offered. Baldwin Locomotive Wks., Testing Equipment Dept, Eddystone, Pa.

Circle No. 23 on Reader Service Card

DELAY LINES are built to specification. Typical unit has 8 pie-section filters, time rise 0.04 microseconds, total delay 0.3 microseconds. Shallcross Mfg Co., Collingdale, Pa.

Circle No. 24 on Reader Service Card

POWDER METAL WHEELS are offered for castor service, from 1¼ to 2½ in. diameters They are self-lubricating, more ductile and lighter. The Bassick Co., Bridgeport 2, Connecticut.

Circle No. 25 on Reader Service Card

RUBBER PIPE, wire reinforced with steel flange backings, cuts noise and vibration. Pressures to 250 psi, temperatures to 150 F taken in 2 to 12 in. pipe sizes. Finn Co., 2850 Eighth Ave., New York 30, N. Y.

Circle No. 26 on Reader Service Card

STAINLESS STEEL BELLOWS are available in complete size range for extreme temperature and corrosion service. Chicago Metal Hose Co., Maywood, Ill.

Circle No. 27 on Reader Service Card

CONTINUED ON PAGE 184

UNIVERSITY OF MICHIGAN LIBRARIES

THE RADIO AND TELEVISION INDUSTRY

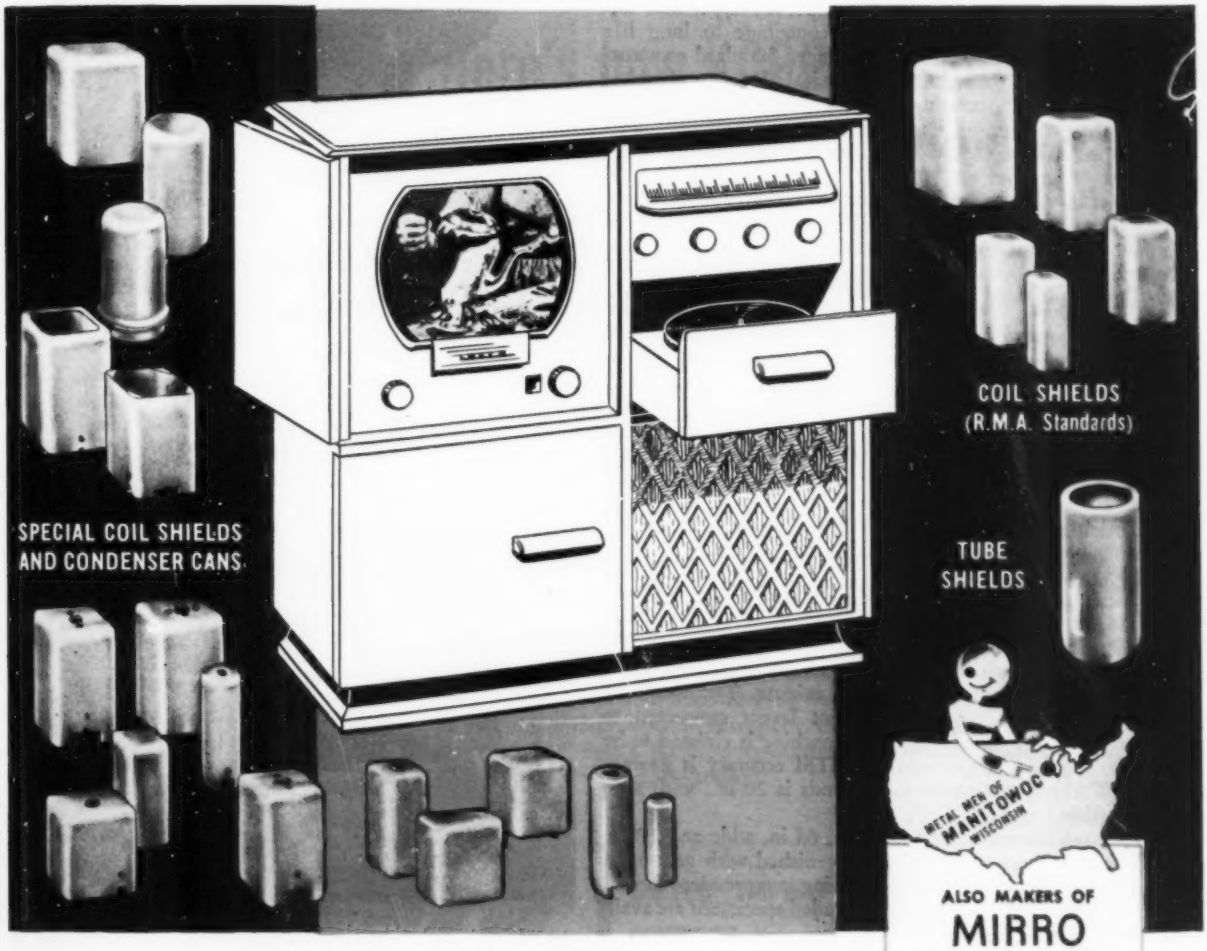
Leading factors in this large field of precision production have regarded us, for many years, as a prime source of aluminum components, manufactured to strict Radio Manufacturers Association standards.

ONE OF THE MANY WE SERVE

WITH ALUMINUM PARTS

Whatever your product may be, if it requires aluminum parts, our diversified skills and facilities can produce them. Alloys and thicknesses? What are your needs? We custom roll sheet metal to your parts specifications, in our own modern rolling mill. Finishes? Satin, color, anodic, polished or chemically brightened. Engineering? Design? Our staffs are at your service.

Your inquiry will focus our full attention on *your* needs . . . at once.



CONTRACT DIVISION

ALUMINUM GOODS Manufacturing Company

MANITOWOC WISCONSIN

FIFTH AVENUE BLDG., NEW YORK 10 MERCHANDISE MART, CHICAGO 54

High Starting Torque Features Single Phase Motor

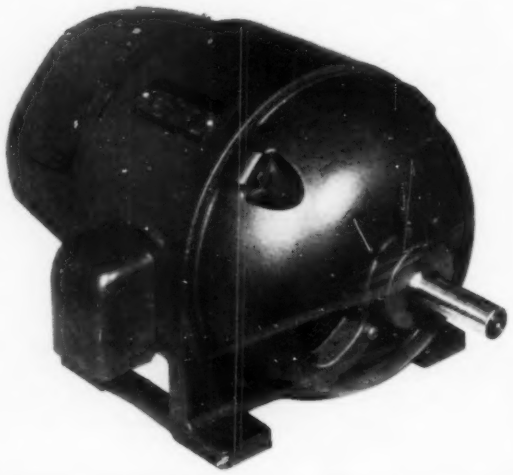
For use where a constant-speed high-torque single-phase motor is required in large ratings, a new repulsion-induction motor is now available. An addition to the manufacturer's standard line, the new motor combines the high starting torque of the repulsion motor with the constant-speed characteristics of the induction motor.

The motor is available in 5-, 7½-, and 10-hp ratings, all 1,800 rpm. The 5-hp unit operates on 115/230 v, while the other two use 230 volts. Typical applications include air and refrigeration compressors, pumps, stokers, floor surfacers, and farm uses such as barn hay curing, wood sawing, silo filling, cold storage and feed grinding.

High starting torque and low starting current are featured, together with positive operation on low voltages for severe starting duty.

The motor is of the open (drip-proof) type, and has been designed with features that contribute to long life and aid in installation and maintenance. Standard construction affords protection against physical damage, electrical breakdown, and operating wear and tear. The cast-iron frame has a damping effect on noise and vibration, and covers are easily removed to facilitate inspection and maintenance of brushes.

To assure good alignment between stator and rotor, the ball bearings are mounted directly in the end shields of the new motor. Improved brush holders give good brush stability, and brushes and rigging are easily serviced. Efficient cooling is provided by a large cast-aluminum fan.



SINGLE PHASE REPULSION-START, INDUCTION MOTOR has been added to the manufacturer's standard motor line. A cast iron frame and protective ventilation screens enclose the drip-proof model. Ratings of 5, 7½ and 10 hp at 1,800 rpm are available. Repulsion poles give high starting torque.

General Electric Co., Schenectady 5, N. Y.

Circle No. 28 on Reader Service Card

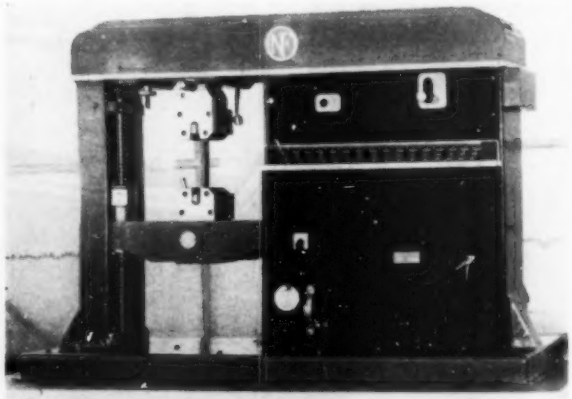
Compact Machine Tests in Three Ranges

A simplified table-model universal testing machine for tension, compression, shear, and transverse tests is available in two capacities: 30,000 lb and 15,000 lb. The 30,000 lb capacity model has additional ranges of 6,000 lb and 600 lb; the 15,000 lb capacity model may be used for 3,000 lb or 300 lb testing.

Loading is mechanical with crosshead speeds infinitely variable from 0.005 to 6 in. per minute. The weighing system consists of knife-edge pivot levers connected to calibrated steel elastic beams. The machine is claimed to be simple, rugged and permanent. ASTM accuracy is guaranteed. Working space between columns is 20 in., with a 14 in. crosshead stroke.

The unit is compact: 42 in. high, 64 in. wide and 18 in. deep, and weighs 1,200 lb. It is furnished with gear-operated wedge grips. Fixtures for making compression, flexure and shear tests and holders for threaded specimens are available as extras.

A cabinet base can be supplied instead of table mounting, if desired. Infinitely variable testing speeds are available. The load scale is linear, minimizing errors in reading and speeding-up operation. A recorder assembly is also furnished, as an extra, to give a permanent record of results. The weighing system is a combination actuating mechanism



with lever and calibrated weight bars. An extra fixture attaches to support columns for transverse testing. The two models available are each adjustable for three capacity ranges. Good accuracy is claimed. Working columns and operating lever are shown in the illustration.

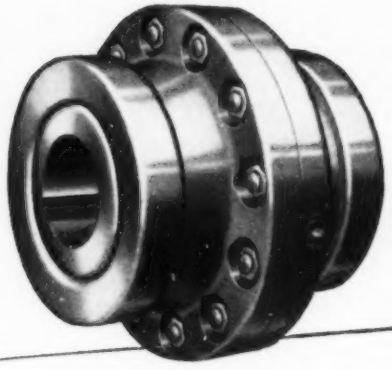
Testing Machine Div. of National Forge & Ordnance Co., Irvin, Pa.

Circle No. 29 on Reader Service Card

CONTINUED ON PAGE 186

GOT A COUPLING PROBLEM

SHEAR PIN?
FLOATING SHAFT?
LOW SPEED?
CUT OUT?
HEAVY DUTY?
REVERSING?
HIGH SPEED?
LIGHT DUTY?



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These FAST'S COUPLING *Services save you money!*

UNSURPASSED ENGINEERING . . . Koppers' engineers are acknowledged the best in the industry. Their practical knowledge, backed with 30 years' coupling experience, is at your service!

IMMEDIATE DELIVERY . . . All standard types and sizes are available for immediate delivery from "on hand" stocks. In case of emergency, just wire factory for special rush delivery!

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No matter what size couplings you need—whatever the service—*see Koppers first!* Because only Koppers offers you Fast's Couplings, *proved* the best insurance for coupling problems you can buy! For 30 years, Fast's have been industry's standard for *low-cost, trouble-free* coupling operation.

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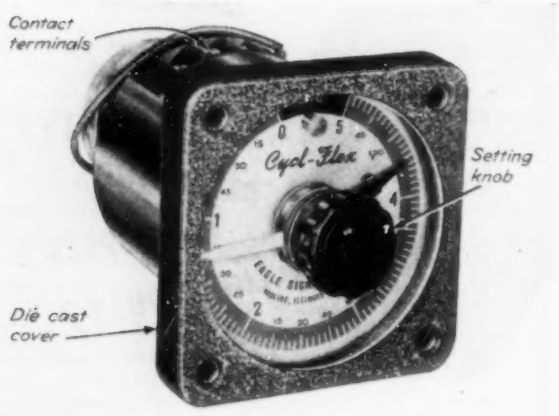
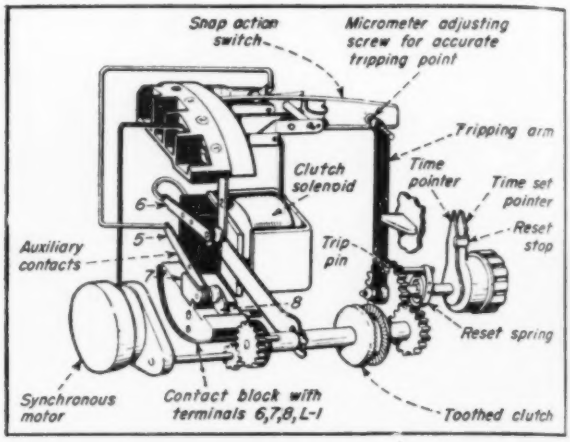
Timer Opens Circuit After Pre-Set Interval



THE TIMER is completely enclosed and mounts in a 3 3/16 diameter hole. An indicating pointer shows the time remaining before the end of the timing period. The dial and pointers are enclosed by an oil-tight die cast cover with a plastic window.



THE LOAD CONTACTS are tripped at the end of the timing interval by a synchronous motor which actuates a tripping arm through a solenoid operated clutch. A spring resets the timer to starting position when the solenoid disengages the clutch.

Three basic models, each available with time scales from 30 sec to 10 hours, provide timers with unusual versatility. Typical applications are dielectric heating equipment, molding presses, photographic printing, chemical processes and similar operations. Load contacts are rated at 10 amps.

One model has two auxiliary contacts mounted on the clutch solenoid. One contact can be used as a holding circuit for applications where the timer is to be started by a momentary contact start switch. The other contact, when used with the snap action switch, provides four possible load circuit combinations. This may eliminate control relays in many circuits. Another model is available without auxiliary contacts. A third model has the clutch action reversed, to reset when the clutch coil is energized. This model is used where a momentary power interruption should not reset the timer or where opening a pilot switch is to start the timer.

Construction features include a slow speed synchronous timing motor and a toothed clutch to eliminate clutch slippage. Dependable operation is further insured by the use of stainless steel shafts and brass cut gears.

Eagle Signal Corp., Moline, Ill.

Circle No. 30 on Reader Service Card

Electronic Timer Has Only One Moving Part

A compact electronic timer which provides automatic control of operation, limit, and sequence timing for industrial processes is now on the market. It is available in three time ranges: 0.06 to 1.2 sec, 6 to 12 sec, and 6 to 120 sec.

Typical applications are: (1) operation timing to control duration of such processes as paint spraying, car washing, photoprinting, electroplating, and heat treating; (2) limit timing to stop conveyor belts if material piles up; and (3) sequency timing with two or more timers in combination to control duration of operations on bearing-grinding machines, rod-coiling machines, commercial potato-peeling machines, centrifuges and illuminated signs.

A detachable back plate mounts directly on a rigid conduit or any flat surface; wires are brought to the back plate and pulled through, and the timer is then mounted on the plate by means of four screws. Wiring to the terminal boards completes the installation. Space between the terminal board and the cover provides room for all front-connected wiring, including that for the accessories. In addition, a dual set of contacts on the timer permits control of two operations on the same time schedule.

The relay armature is the only moving part, keeping mechanical wear low. The electronic tube is a standard model. The timer operates on 115-130 v, 50-60 cycle current. The entire unit measures 8¼ in. high, by 5½ in. wide, 4 in. deep.

Tests indicate that the new timer can perform a million or more operations at these controlled load requirements: inrush, 15 amps; carry, 10 amps, and break, 5 amps. As the controlled load demand is reduced, more operations are expected. Construction consists of one tube, a timing potentiometer calibrated in seconds on a locking type knob, a linear timing scale, an accessible wire terminal board, and a relay with 2 SPDT form "C" contacts.

General Electric Co., Schenectady 5, N. Y.

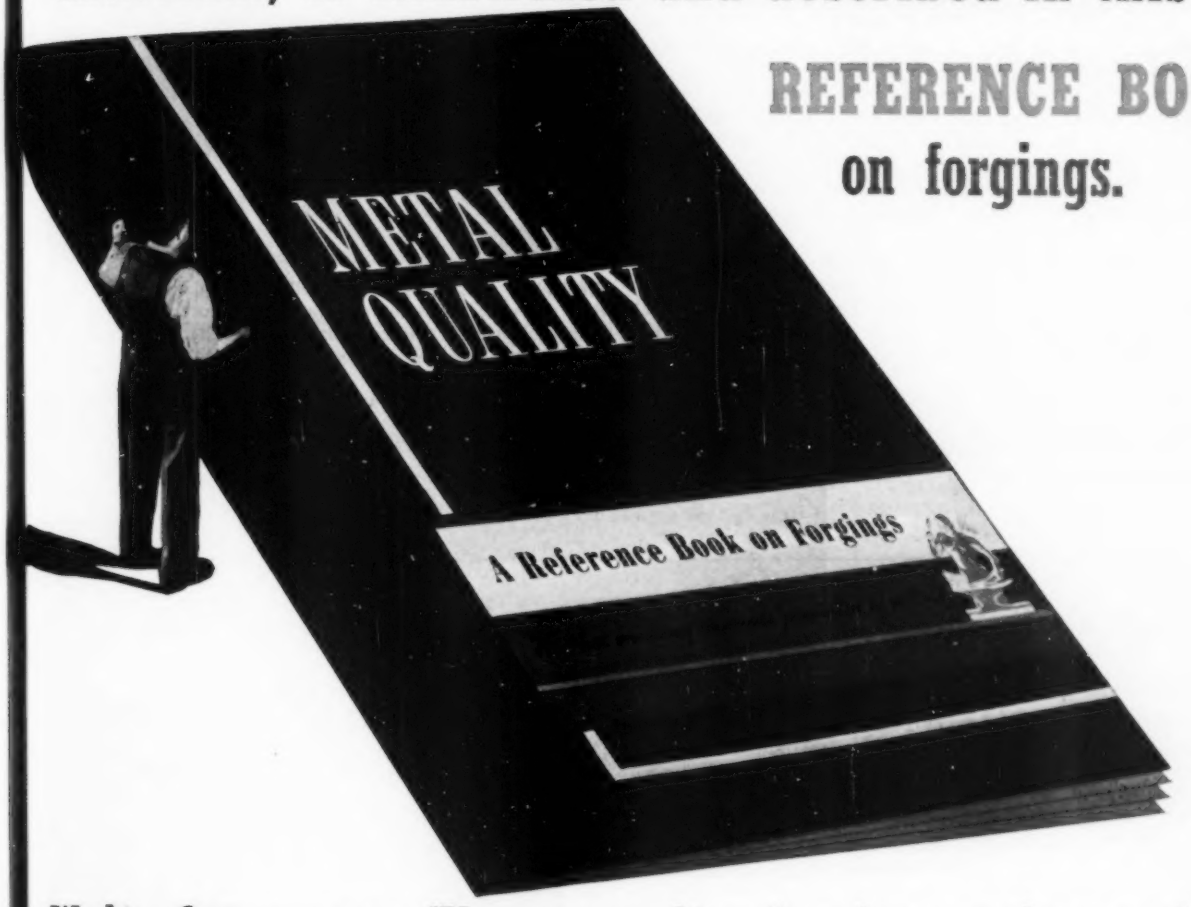
Circle No. 31 on Reader Service Card

CONTINUED ON PAGE 188

UNIVERSITY OF MICHIGAN LIBRARIES

What *Forgings* have that offer so many more economic, engineering and production advantages than can be obtained with any other material, is illustrated and described in this

REFERENCE BOOK on forgings.



Write for a copy. Then consult a forging engineer about how you can obtain the correct combination of mechanical qualities in forgings for your particular type of equipment.

DROP FORGING ASSOCIATION
605 HANNA BUILDING • CLEVELAND 15, OHIO

DROP FORGING ASSOCIATION • 605 Hanna Building • Cleveland 15, Ohio

Please send 60-page booklet entitled "Metal Quality—How Hot Working Improves Properties of Metals", 1949 edition.

Name..... Position.....

Company.....

Address.....

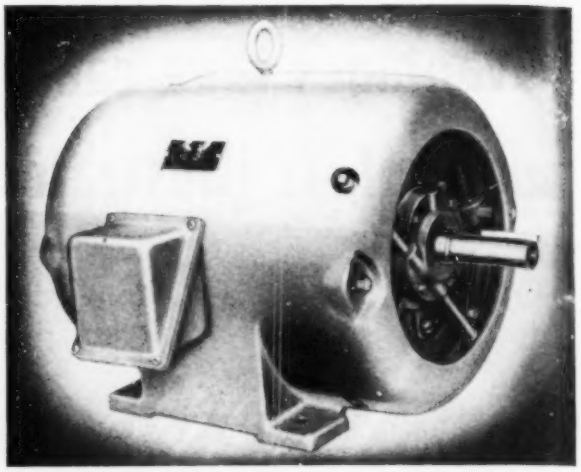
Cast Iron Frames Protect Outdoor Motors

A full line of cast iron frame totally enclosed fan-cooled motors is available in ratings from 5 to 250 hp in both standard and explosion-proof designs. These motors are suitable for applications in chemical plants, oil fields, petroleum refineries and similar plants where corrosion is a problem, either because of the nature of the materials processed or because the motors must be installed outdoors.

The inner frame, outer frame and feet are cast in one piece. End plates are also cast iron, ribbed for strength. Cap screws secure them in the correct position, as determined by matched machined surfaces on frame and end plates. A removable blower shield with integral air intake protects the external blower. A cast iron shield on the drive end deflects the emerging air stream around the bearing. Blowers are malleable iron on standard motors and non-sparking bronze on the explosion-proof models. On all except the largest ratings, blowers are designed to permit rotation in either direction.

The diagonally split conduit box bolts to a nipple which is brazed to the stator frame. Corrosion-resistant cap screws and gasket assemble with the box. Leads are brought out through the nipple which is filled with a flexible sealing compound. Stator laminations are isolated from the air stream by the cast iron case.

After the windings are in place, the stator is baked to remove moisture, and while still hot, immersed in a heavy insulating varnish. Complete impregnation of the windings results from the suction created by dipping the hot rotor into cooler varnish. The coat is baked dry and the process is repeated. A final coat of special acid- and alakli-resistant material gives a tough, oil-proof coating.



BEARINGS ARE PRELUBRICATED and can be re-lubricated when necessary. Pressure cast aluminum rotor is dynamically balanced. Corrosion resistant motors from 5 to 250 hp are available totally enclosed or explosion proof.

Wagner Electric Corp., 6400 Plymouth Ave., St. Louis 14, Mo.

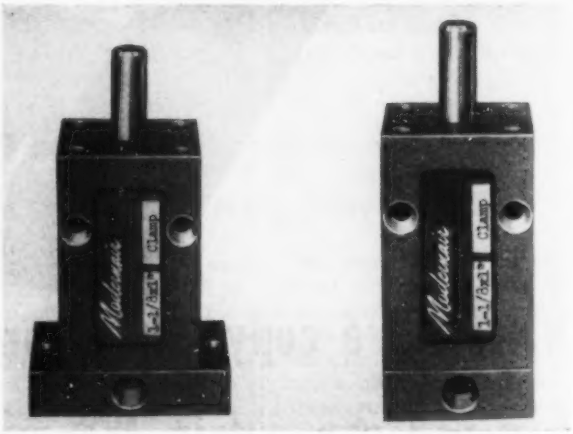
Circle No. 32 on Reader Service Card

Single-Acting Square Air Cylinders Mount Closer Together

Square body design is now being used in air cylinders, primarily to save space in air clamp installations. Three bore sizes, 1½, 2 and 3 in., are available, single acting, spring return.

The square actuating cylinders can be mounted closer together than round units. Bodies and caps are aluminum extruded stock. The cap may be mounted to face the inlet port in the direction most convenient for piping connections. A long piston guide bushing and close piston and bushing tolerances are among manufacturer's claims. Piston seal is a standard "O" ring.

The plain end-cap model for horizontal mounting is shown at right in the illustration. The end cap also serves as a mounting flange on the second model. The forward faces of both cylinders are tapped for front attachment.



Modernair Corp., 4222 Hollis St., Oakland 8, Calif.

Circle No. 33 on Reader Service Card

HIGH-Q COILS for sub-audio frequencies have stable characteristics for 0.1 to 10 cycles. United Transformer Co., 150 Varick St., New York 13, New York.

Circle No. 34 on Reader Service Card

VALVE SEAT INSERT is claimed free of checking and burning at high temperature. Dissipates heat and does not loosen. Martin Wells, 5886 Compton Ave., Los Angeles 1, Calif.

Circle No. 35 on Reader Service Card

CRASH SWITCH cuts off airplane electrical equipment by inertia on 3-G deceleration. Reset button provided. Simmonds Aerocessories, Inc., Tarrytown, N. Y.

Circle No. 36 on Reader Service Card

CONTINUED ON PAGE 190

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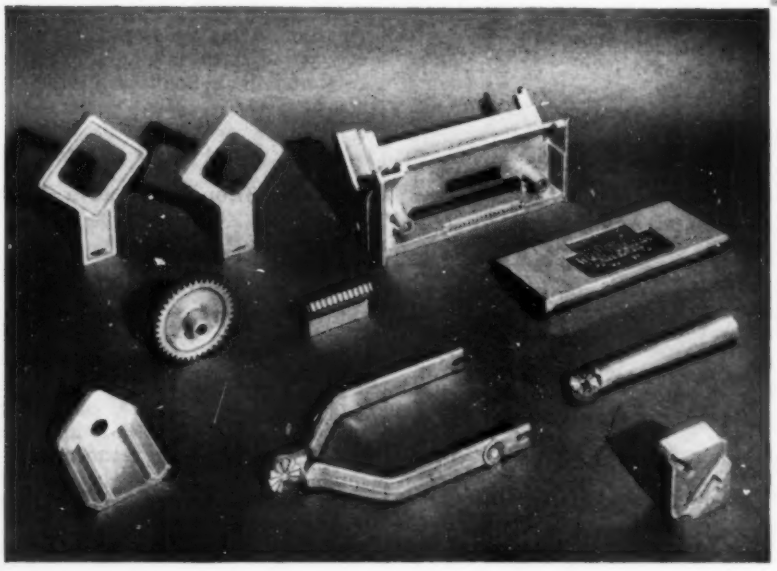
To produce a Superior Food Cutter at a Marketable Price

PERMITE ALUMINUM CASTINGS were selected

The Model "K" French Fry Cutter, manufactured by the H. G. W. Young Company of Philadelphia, affords all the advantages of light weight, easy cleaning, non-toxicity. It is rustproof, corrosion-proof. Here's why Permite Permanent Mold Aluminum Castings (shown below) were used for 10 important parts, as told by the Manufacturer.



HIS LETTER SAYS:

". . aluminum was selected . . because of the non-toxicity and lightweight characteristics of this metal. However, more important than this is the non-corrosive nature of aluminum . . none of the parts will rust or corrode. . ."

"In addition . . we were fortunate to find that with aluminum permanent mold castings, due to the small amount of machining time required, we were able to put the unit out at a price which is marketable."

One of the easiest of all metals to work, non-corrosive, non-toxic, lightweight—Permite Aluminum Castings, with a precision tolerance as close as ±.010", could easily be the answer YOU are looking for. Write us about your parts problem, or send drawings for recommendations and estimate.

80-page manual on the design, production and uses of aluminum castings sent on request.

PERMITE

ALUMINUM INDUSTRIES, INC.

CINCINNATI 25, OHIO

DETROIT: 899 New Center Building. NEW YORK: 9 Rockefeller Plaza. CHICAGO: 64 E. Jackson Boulevard. ATLANTA: 423 Grant Building.

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PRECISION MOLDERS
62 William Street, New York 5, N. Y.
Digby 4-0780

NEWS on NYLON Molding

Our new molding technique permits us to state unequivocally that:

1. Nylon parts cost less (often considerably less) than metallic components they replace in the majority of cases.
2. Die costs are not expensive.
3. Small runs are possible at a reasonable price.
4. Our prices on large runs are definitely competitive.

We offer the assistance of our research engineers to help solve your problems in connection with Nylon Molding.

If you are not now using Nylon, our product study service is available. Possibly we can show you how advantageous Nylon Molding can be with our methods and equipment.

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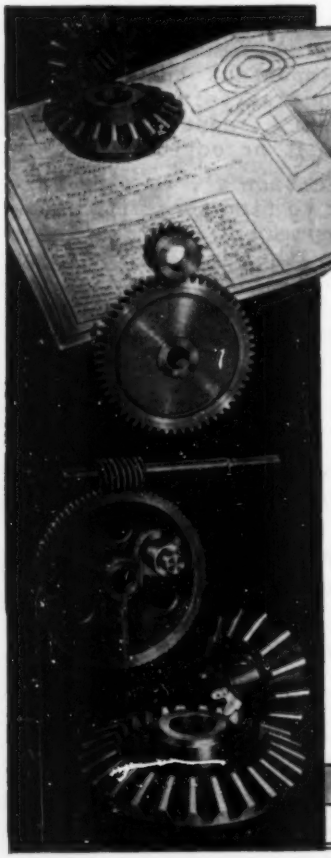

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Precision Gears Made to Your Exacting Specifications

When Sewall craftsmen go to work on your specifications, you are employing the extensive facilities of an experienced organization. Our engineers can give you valuable assistance in establishing the *one best design*—then in turning out uniform production runs. Illustrated are examples of a Sewall bevel gear and pinion . . . worm and worm gear . . . helical gear and pinion . . . Zerol bevel gear and pinion. *Send blueprints or samples for prompt estimates on any quantity.*

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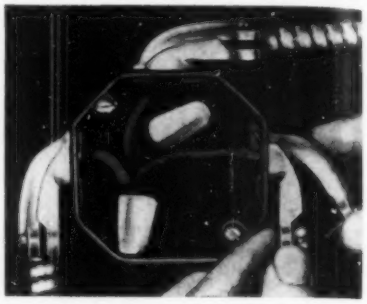
RACKS • SPROCKETS • SPUR • SPIRAL • BEVEL • ZEROL BEVEL • WORM GEARS

E. B. SEWALL MANUFACTURING CO.
696 Glendale St. St. Paul 4, Minnesota.

Cable Connectors Snap Into Casting Holes

A new series of $\frac{3}{8}$ in. connectors attach armored electrical cable to $\frac{7}{8}$ in.



cored holes in castings. The die cast aluminum fittings are used with 3/64 to 7/32 in. wall thicknesses.

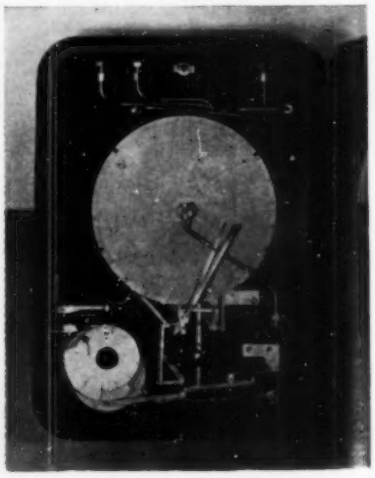
Two claws engaging the inside edge of the hole are spread by a cam-wedge action when the machine screws are tightened. This locks the cable to the casting. Only a screw driver and a $\frac{7}{8}$ in. cored hole are needed for installation. There are no threads, washers, wrenches or locknuts. One connector fits all applicable wall thickness.

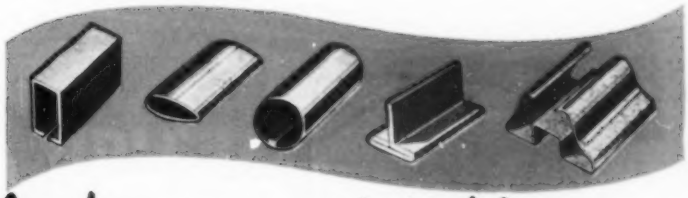
Unimatic Corp.,
52 E. Center St., Nutley, N. J.

Circle No. 37 on Reader Service Card

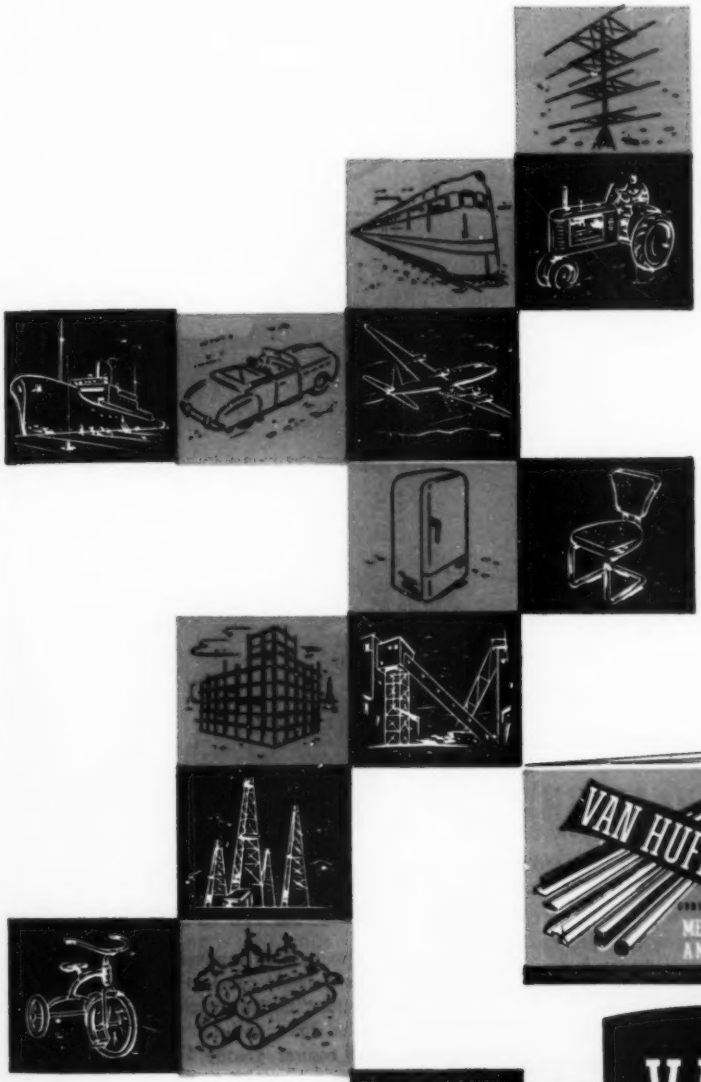
Program Recorder-Controller Eliminates Cam Cutting

This instrument is designed to give repetition of a predetermined control program with relation to temperature or pressure. There are no cams to be cut. Standard cams are set to the conditions required and can be changed when necessary. It is possible to change the rate rise, the length of hold, and the holding temperature or





VAN HUFFEL metal shapes and tubing..



are proving their value in these widely diversified industries

Whatever your product .... battleships or baby buggies .... toys or television equipment .... it's just possible that Van Huffel rolled shapes or tubing can simplify production and lower costs.

You will find, too, that Van Huffel's "under one roof" procedure—designing and building forming dies *plus* cold forming shapes and tubing in continuous lengths—is worth your consideration.

Illustrated brochure gives you the whole story. Write for it today.

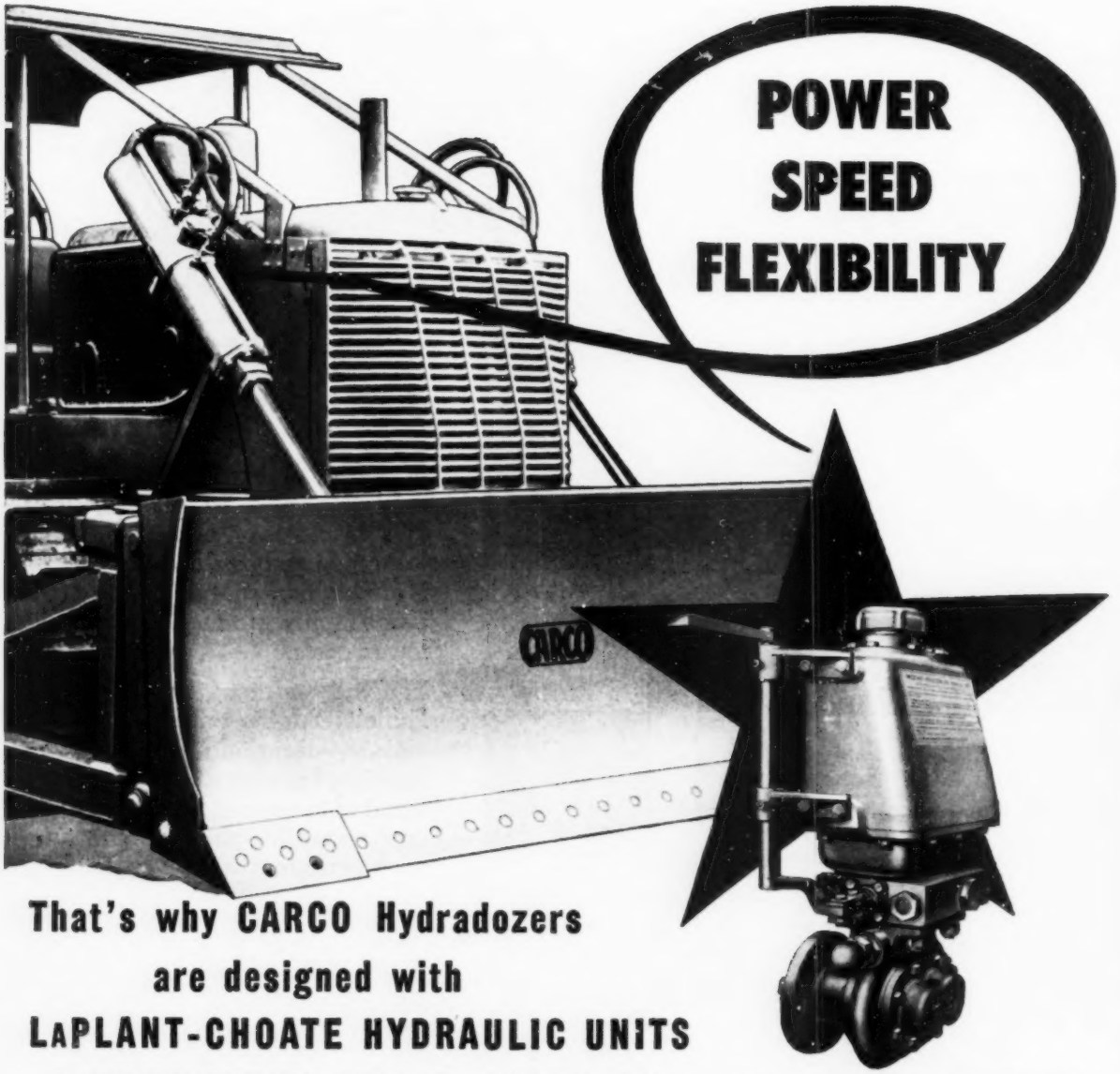
Van Huffel forms to your specifications: Welded, Lock Seam, Open Seam and Butted Tubing; Shapes and mouldings in any metal in thicknesses up to 5/16".

HOW ABOUT Yours?

VAN HUFFEL
TUBE CORPORATION · WARREN · OHIO

continuous rolled

METAL SHAPES · TUBING



That's why CARCO Hydradozers are designed with LaPLANT-CHOATE HYDRAULIC UNITS

LAPLANT-CHOATE Hydraulic Units put muscle into the equipment that has to slug it out with hard digging and heavy loads — they supply the speed and flexibility that keeps the job moving economically and profitably. That's why so many manufacturers like Carco use LPC Hydraulic Components — jacks and the all-in-one fluid power control — to provide maximum power, operating smoothness and precision, instant response to controls.

DESIGN YOUR PRODUCT FOR BETTER PERFORMANCE

Full hydraulic operation of your product will give you sales advantages that keep you well ahead of competition— when you specify LaPlant-Choate Hydraulic Units. Proved by more than 25 years use as standard equipment on LPC earthmoving equipment, LPC Hydraulic Units assure you of top flight efficiency and economy.

The Famous Single-Unit LPC Fluid Power Control

Pump, valve and oil reservoir are all combined into a single compact unit. This new "closed system" design completely eliminates long suction lines, reduces the hazard of leaks and assures greatly increased speed and efficiency. In addition, the entire unit is skillfully engineered and precision built for utmost simplicity, easy servicing, and long, trouble-free operation.

Ask for Bulletin B-1152B. LaPlant-Choate Manufacturing Co., Inc., Cedar Rapids, Iowa.

LaPLANT CHOATE

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pressure without cutting a new cam for each cycle. The holding temperature is adjusted by a control knob. The recorder has a 9 in. chart and a standard 12 in. case.

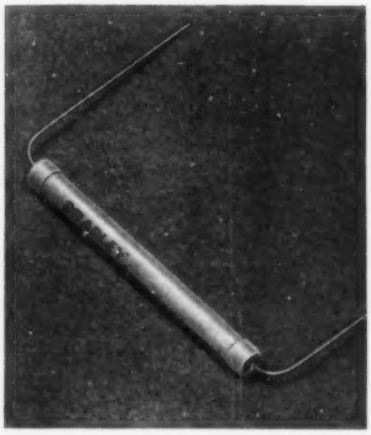
Temperature and pressure systems are interchangeable. The instrument can be modified to control two variables simultaneously, to include automatic reset after cycle completion, to stop the cam at any critical point, or to provide for the addition of signalling devices or the use of air or electric relays or switches.

Gotham Instruments Div., American Machine and Metals, Inc., 149 Wooster St., New York 12, N. Y.

Circle No. 38 on Reader Service Card

Selenium Rectifier For Low Currents

A low current selenium rectifier with a special plate for use on d-c micro-amp currents has been developed to provide an efficient rectifier for very low current requirements. It is rated up to 6,000 v peaks inverse and is connected in series for higher voltages. Current rating is 200 amps. The rectifier is $\frac{1}{4}$ in. diameter by 3 in. long.



Bradley Laboratories, 82 Meadow St., New Haven, Conn.

Circle No. 39 on Reader Service Card

Throttle Controls Automatic Clutch

Designed for use on the manufacturer's 3.0-4.5 and 5.2 hp air cooled engines, this variable transmission operates through an automatic clutch which is controlled by the throttle to give speed ratios ranging from 3.1 to 0.95:1. A clutch is mounted on the engine drive shaft and connects to the driven member by V-belt. This driven

SPECIFY PARKER
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THE J.I.C.
TUBE FITTING

- 37° flare angle
- Dryseal pipe threads
- has *actually* passed rigid AN-F-47 performance test as required by J. I. C. standards. *Certified laboratory test report* available.

MADE BY THE WORLD'S LARGEST MANUFACTURER OF LEAKPROOF TUBE FITTINGS.

- precision-made to be *leakproof* under even the severest operating conditions.
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- 3-piece design—remains *leakproof* after repeated reassembly.
- accommodates full range of standard tube wall thicknesses.
- made in steel, stainless steel, also aluminum alloy.
- shape bodies machined from forgings, other parts from bar stock.
- stocked by distributors across the nation.
- write for Parker "Industry Standard" Tube Fittings Catalog 203.

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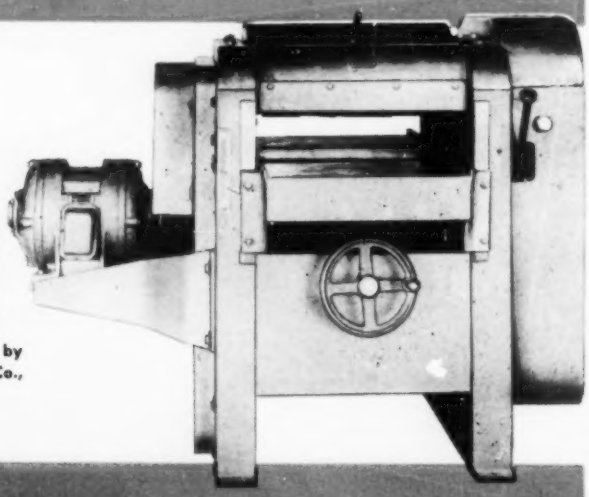
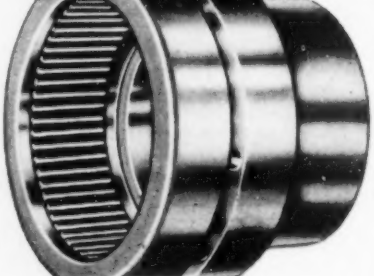
POWERMATIC THICKNESS PLANER

is a heavy duty surfacer used in woodworking for production planing and to produce a precision finish required by cabinet and pattern shops. Manufactured by Powermatic Machine Co., McMinnville, Tenn.



Gains 4 Important Advantages
by replacing plain bearings with
ORANGE ROLLER BUSHINGS

1. Solves limited space problem within bearing housings.
2. Eliminates hardening necks of idler and pressure rollers.
3. Eliminates former periodic bearing replacements and expensive roller replacements. Over two years using Orange Roller Bushings and still no replacements.
4. Obtains smooth-running, precision bearings which operate under varying pressures with little or no wear, under abusive conditions.



YOU can depend on Orange Roller Bushings for trouble-free, anti-friction protection of rotating or oscillating parts—large or small—under heavy loads or high speeds—in every kind of product from precision machine tools to heaviest equipment.

Orange Roller Bushings have high load carrying capacity in small diameters—run smoothly, quietly—provide long service life. Closer internal running clearances minimize possibility of mis-alignment of rollers. All raceways are through-ground.

Full range of stock sizes, from ½" to 8" shaft diameters. Write for Engineering Data Book.

ORANGE ROLLER BEARING CO., INC.
551 Main Street, Orange, N. J.

member is mounted on a jackshaft or on a sliding gear transmission to give an additional low speed of 2.25:1 and reverse. Belt tension is controlled by the action of springs on a cam surface in the driven member. The power units are designed for materials handling equipment and vehicles requiring simplicity and smooth operation.

Cushman Motor Works, Inc., Lincoln, Neb.

Circle No. 40 on Reader Service Card

Smaller Tire Lightens Load

A 4.00 by 18 pneumatic tire disk wheel has been designed 10 lb lighter than the 6.00 x 16 tire model. A drop center rim holds the tire. The disk is formed by two plates which are bulged for strength and lock to the seamless steel hub. Either tapered or straight roller bearings are used in the 4½ in. long hub. With a diameter of 27 in. (about equal to that of 6.00 x 16 automotive) these tires are designed for good cushioning.

Spindle sizes are 1.190, 1¼, 1 5/16 and 1⅜ in. on tapered and 1¼ on straight roller bearings. 2 and 4 ply tires take rated loads of 1,000 and 1,575 lb at 5 mph and 835 and 1,310 lb at 10 mph respectively.

Geneva Wheel Co., 1013 Railroad St., Geneva, Ohio

Circle No. 41 on Reader Service Card

One Revolution Per Week Motor Stands 1⅜ in. High

An extra-slow-speed timing motor is available in standard speeds of 6, 8, 12 hours, 1 day and 1 week per revolution. On quantity orders non-standard speeds are offered in the ranges of 5, 9, 10, 15, 16, 30, 32 and 36 hours, and 2, 4, 10 and 30 days per revolution.

The new series of timing motors has been developed to provide a lower cost, compact unit capable of operating at very low speeds. The manufacturer has attempted to meet the objections of excessive bulk and comparatively high cost raised to previous slow speed timers. Lubrication has been simplified and the number of components has been reduced in an effort to lower manufacturing costs. Overall depth of the unit has been reduced to 1⅜ in. through refinements in gear cup and gear train design. The motors are interchangeable with faster speed motors in mounting and other dimensions. Typical applications are in day-night thermostats, daily program

timers and refrigerator defrosting mechanisms.

Haydon Mfg. Co., Inc., Torrington, Conn.

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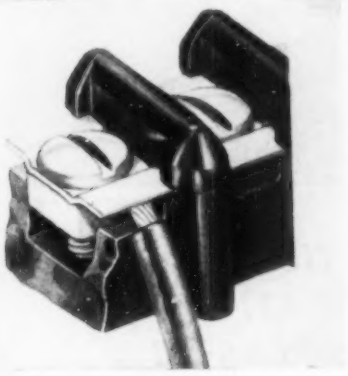
Terminal Clamps Wires

A new terminal block connector clamps two wires under a single screw. There are no lugs, wire loops or wire preparation. The body of the terminal block is molded in solid form from



general purpose thermo-setting material. Screws, nesting in a blind hole, cannot short to ground. Blocks with from 1 to 22 terminals are available. A maximum wire size of AWG 18 stranded or 16 solid is handled.

Curtis Development & Mfg. Co., 3266 N. 33rd St., Milwaukee 16, Wis.

Circle No. 43 on Reader Service Card

Tiny Tube Capacitors

A miniature capacitor has been designed for severe service. The size reduction is attained primarily by a metallized-paper section which is impregnated and placed in a non-magnetic hermetically-sealed metal case with vitrified ceramic terminal seals. Operating range is —55 C to +50 C with temperatures to 95 C possible with voltage derating. Power factor is less than 1 percent at a frequency of 1000 cps and ambient temperature of 25 C.

Claimed to be the world's smallest standard capacitors, the new units are available in 200, 400 and 600 v d-c, 0.000 to 2.0 mfd capacitances. Dimensions range from 0.175 in. dia by 7/16 in. long, up to 0.670 in. by 27/32 in. long. Optional plastic insulating sleeves add 0.062 in. to the diameter and $\frac{1}{16}$ in. to length.

Aerovox Corp., New Bedford, Mass.

Circle No. 44 on Reader Service Card

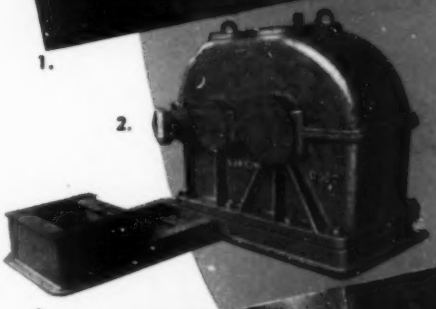
INSTALLATIONS LIKE THESE YOUR ASSURANCE OF "GEARABILITY"

Lufkin

Lufkin Gear Speed Reducers and Speed Increasers are dependable performers in industry's toughest services. A few installations are shown here. Couch and Press Sections of Newsprint Machines, Refineries, Chemical Plants and the Oil Field number among other Lufkin installations where stamina and ability are so important. Lufkin gear specialization can be of particular value to you now as you plan for the most in cost saving, outstanding equipment operation.

Lufkin "advanced engineering" gives you horizontal parallel shaft Herringbone gear speed Reducers and Increasers for every application. Though your problem may be unusual, the Lufkin representative nearest you, or the home office, will put Lufkin experience to work for you to help you achieve the excellence you want.

1.



2.

3.

1. Lufkin No. 2412 Speed Increaser, driven by 350 h.p. Diesel engine in pipe line pumping service.
2. Large double reduction Gear Reducer with motor base for Georgia paper mill.
3. Special 400 h.p. Speed Increaser driving centrifugal pump at 3500 r.p.m. and driven by 400 r.p.m. engine.

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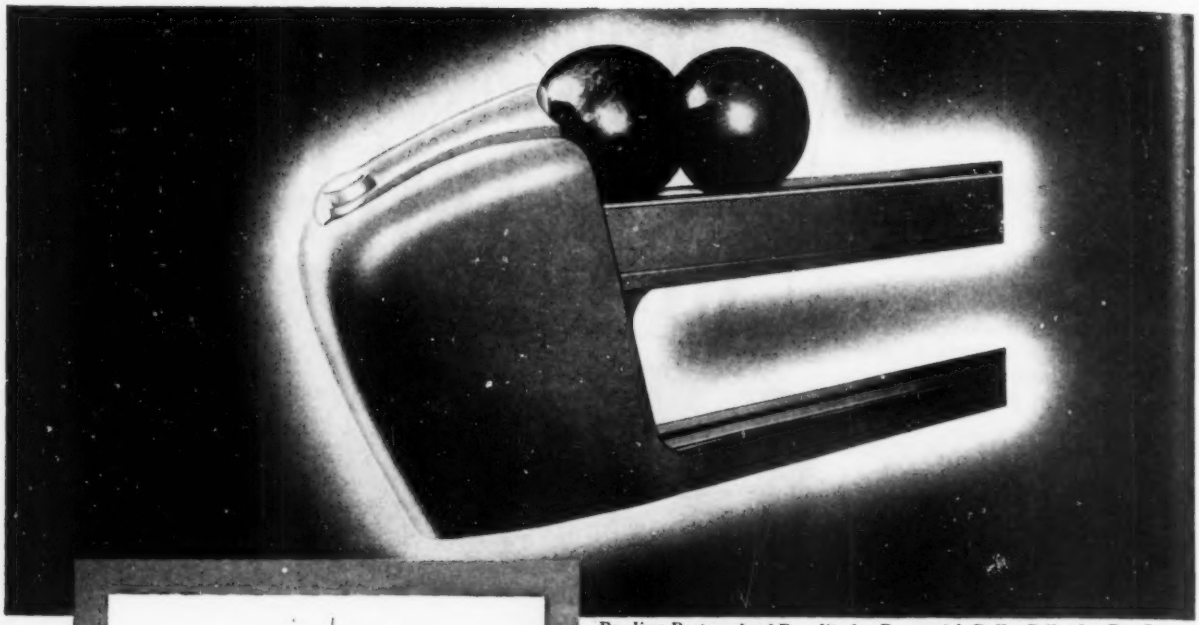

Bowling Post made of Royalite for Brunswick-Balke-Collender Co., Inc.

Royalite

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FUNCTIONAL, DECORATIVE THERMOPLASTIC

There is literally no end to the uses of U. S. Royalite, the thermoplastic that dozens of manufacturers use for a wide variety of products.

This remarkable material opens up new design possibilities—either in designing *new* products, or improving your present product.

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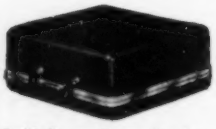
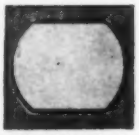
Friendly to Touch	Sanded—Punched	Hard or Soft
Quiet and Warm	Nailed	Coarse or Fine-Grained
Relatively Light	Bolted—Riveted	Deep-Grained or Shallow
Shatterproof	Sewed	Thick 1/2" to Thin 1/32"
Stainproof	Screwed Into	Glossy or Dull
Acid Resistant	Cemented	Decorated or Plain
Excellent Dielectrically	Buffed—Polished	Bright or Somber Colors

*Tote Box

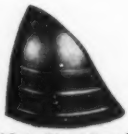


*Smith-Corona Typewriter Case



*Television Mask

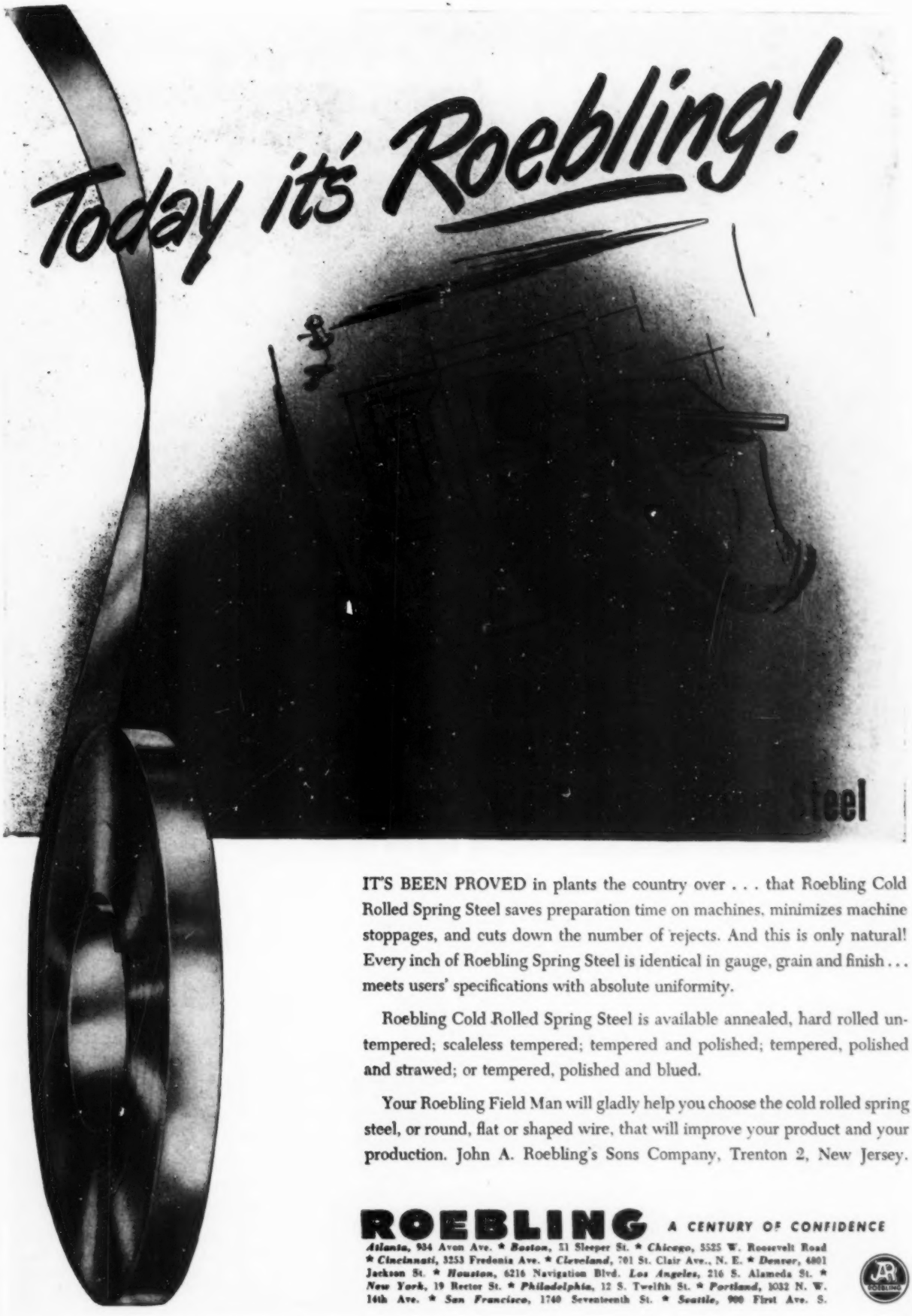
*Home Freezer Breaker Strips



*Auto Stone Guard

UNITED STATES RUBBER COMPANY
2638 North Pulaski Road, Chicago 39, Illinois

U.S. Royalite



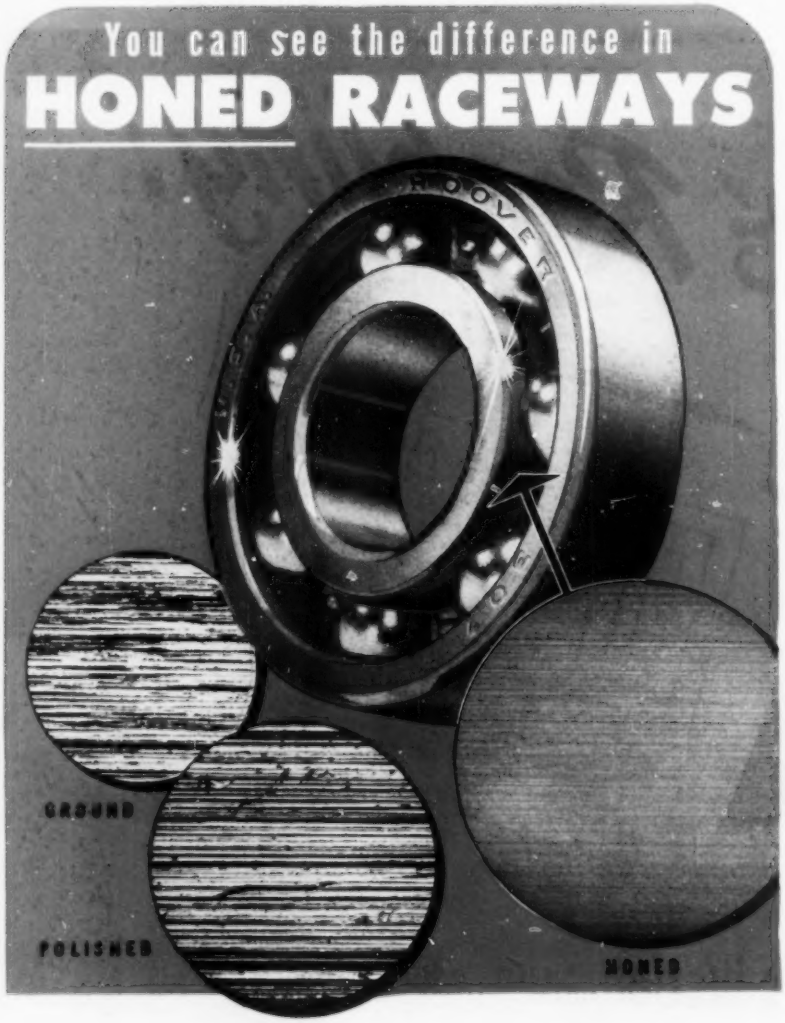
IT'S BEEN PROVED in plants the country over . . . that Roebling Cold Rolled Spring Steel saves preparation time on machines, minimizes machine stoppages, and cuts down the number of rejects. And this is only natural! Every inch of Roebling Spring Steel is identical in gauge, grain and finish . . . meets users' specifications with absolute uniformity.

Roebling Cold Rolled Spring Steel is available annealed, hard rolled untempered; scaleless tempered; tempered and polished; tempered, polished and strawed; or tempered, polished and blued.

Your Roebling Field Man will gladly help you choose the cold rolled spring steel, or round, flat or shaped wire, that will improve your product and your production. John A. Roebling's Sons Company, Trenton 2, New Jersey.

ROEBLING *A CENTURY OF CONFIDENCE*

Atlanta, 934 Avon Ave. ★ *Boston,* 11 Sleeper St. ★ *Chicago,* 5525 W. Roosevelt Road ★ *Cincinnati,* 3253 Fredonia Ave. ★ *Cleveland,* 701 St. Clair Ave., N. E. ★ *Denver,* 4801 Jackson St. ★ *Houston,* 6216 Navigation Blvd. *Los Angeles,* 216 S. Alameda St. ★ *New York,* 19 Rector St. ★ *Philadelphia,* 12 S. Twelfth St. ★ *Portland,* 1032 N. W. 14th Ave. ★ *San Francisco,* 1740 Seventeenth St. ★ *Seattle,* 900 First Ave. S.



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HOOVER BALL BEARINGS

The photographs reproduced above are magnified one hundred times so you can see the difference between ground, polished and honed raceways. The process and the special machines for the honing operation are exclusive, patented, Hoover developments. Honing by Hoover goes far beyond grinding and polishing to achieve a surface that assures amazing quietness plus *90 per cent longer bearing life . . . 30 per cent greater load capacity.* That's why Hoover Ball Bearings are the choice of distinguished American manufacturers of fine machines and equipment.

The Hoover Engineering Manual will be mailed free to engineering and purchasing executives requesting a copy on their business letterhead.

THE ARISTOCRAT OF BEARINGS

HOOVER BALL and BEARING CO.
Ann Arbor, Michigan

OUR READERS SAY

Designing, Not Refining

To the Editor:

The article, "Beryllium's Toxicity Is Largely Myth", PRODUCT ENGINEERING, September 1949, page 155, attempted to convey the impression that beryllium is no more toxic than copper, for example, and presented a fascinating but wholly misleading picture of the situation. While the author states that the article is based upon facts, he has actually completely ignored the vast majority of medical literature on the subject of beryllium. The whole tenor of the article is summed up in the conclusions, one of which states that "An acid radical in connection with beryllium is necessary to cause either of the two forms of pneumonitis". This is simply not consistent with the facts, even as presented in the article. It is pointed out that many cases occurred in the fluorescent lamp industry, yet there is no acid radical involved. In addition, much stress is given the absence of cases in processing low beryllium alloys. This is true, but does not justify the statement that "Beryllium Toxicity Is Largely a Myth". In order to judge this statement in its proper light one should examine all of the facts.

There have been reported in the medical literature nearly 400 cases of acute pneumonitis and over 100 cases of chronic pneumonitis in such widely different industries as fluorescent lamp, sign tubing and fluorescent powder manufacturing; production of ceramics involving the use of beryllium oxide; heat treating, rolling and machining beryllium metal; manufacture of beryllium alloys (involving melting 4 percent master alloy). With regard to the last, Dr. Leroy Gardner, late Director of the Saranac Laboratory, cited nine pulmonary deaths among workers in a brass foundry casting beryllium alloys, in his paper "Generalized Pulmonary Granulomatosis Occurring Among Workers Believed to be Exposed to Beryllium or its Compounds", which was delivered at a meeting of the Industrial Hygiene Foundation in Pittsburgh in November 1946.

There has also been published by the United States Atomic Energy Commission Bulletin AECD-1803 entitled "Beryllium Symposium" which con-

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tains an excellent summary of the problem including detailed descriptions of cases. Those who are interested in this problem should make it a point to study this report as well as the work of Van Ordstrand, Hardy, Martland, Machle and many others.

It is a significant fact that a study of the distribution of beryllium disease indicates that beryllium toxicity is far from being a myth. It is significant, first, that several compounds of beryllium are capable of producing this disease; that cases have been reported from a variety of widely scattered industrial operations involving the use of several different compounds of beryllium; and finally the only common industrial exposure of the persons involved has been to beryllium compounds.

It is thus obvious that any manufacturer using or contemplating the use of any material containing beryllium should investigate this problem thoroughly before reaching the conclusion that he is handling or about to handle an innocuous material.

F. H. STERNER, M.D., Consultant, Industrial Health, U. S. Atomic Energy Commission; K. P. DUBOISE, Assistant Professor of Pharmacology, University of Chicago; A. A. SANDER, M.D., Associate in Medicine, Marquette University Medical School; T. L. SHIPMAN, M.D., Health Division Leader, Los Alamos Scientific Laboratory; P. D. WIKES, Professor of Industrial Hygiene, Harvard School of Public Health; R. H. KEHOE, M.D., Professor of Industrial Medicine and Director of Kettering Laboratory, University of Cincinnati; H. C. HODGE, Professor of Pharmacology, University of Rochester School of Medicine and Dentistry; C. R. WILLIAMS, Assistant Professor of Industrial Hygiene, Harvard School of Public Health; H. L. HARDY, M.D., Assistant Medical Director in Charge of Occupational Medicine, Massachusetts Institute of Technology.

Ed—We are in complete sympathy with the views taken by the above medical authorities, but believe that, in the light of the problems faced by our design-engineer readers, we gave them an honest answer.

When we say "Beryllium Toxicity Is Largely Myth" we are speaking of the industrial alloys that many of our readers know and have been using for years. These engineers are not concerned with mining, smelting and alloying—they want to know whether beryllium alloys are safe to use as a standard fabricating material. We believe these standard alloys (which have been in use at least 15-20 years) to be safe and useful materials.

Medical literature was not ignored in our search for the facts. The medical files of the Metropolitan Insurance Company were studied. Unfortunately we found that most medical research

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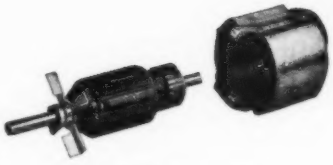

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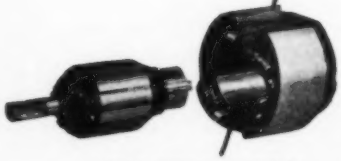

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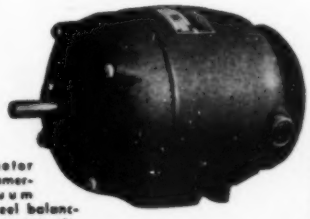
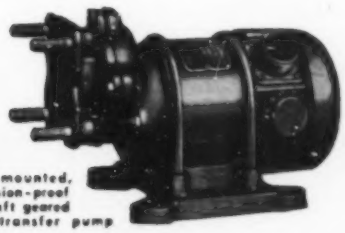


Universal motor used for commercial vacuum cleaners, wheel balancers, air compressors, etc.



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dealt with the mining and smelting of the basic ore, or with beryllium compounds used in fluorescent lighting—not with the low alloy strip and wire about which our readers want to know. Before the article went to the printer, copies were sent to all people furnishing information. In addition, we sent a copy to the American Medical Association, in Chicago. We asked the AMA to comment on the article and make any changes in fact that they thought advisable. The AMA did not bother to answer our letter or return the manuscript.

Crossing the Delaware

To the Editor:

We would like to refer you to the "Where Can I Get It" column on page 206 in the July 1950 issue of PRODUCT ENGINEERING. In answer to E. F. Bruning's query on Fluorine Seals, you referred him to our Company as a moulder of "V" packings. While we certainly are appreciative of this publicity, we do not appreciate our sudden removal to Camden, N. J. As you can see by our letterhead, we are very much alive and kicking still in Philadelphia. —S. K. AITKIN
Linear Incorporated

Ed—We apologize to Mr. Aitkin for mislocating his company on the wrong bank of the Delaware River. Linear's address is, of course, State Road & Levick Street, Philadelphia 35, Pennsylvania.

Engineers in Management

To the Editor:

The July issue of PRODUCT ENGINEERING has just arrived and, as always, I read the editorial first. These discussions usually "hit me just right" and the *Cloth of Gold* especially, because we need more engineers not only in sales, but in all phases of manufacturing. The more technical the product, ours is vacuum tubes, the more engineers.

As an extension of the editorial, I do not mean more engineers in a bigger engineering department. We need more engineers as foremen and supervisors. Many of our industries and plants are still full of school teachers, shoe salesmen and farmers that entered industry in 1941 and 1942 and are doing a fair job. However, they sometimes impede progress by their lack of engineering training and insight. The serious thing about this situation is that industry is requiring

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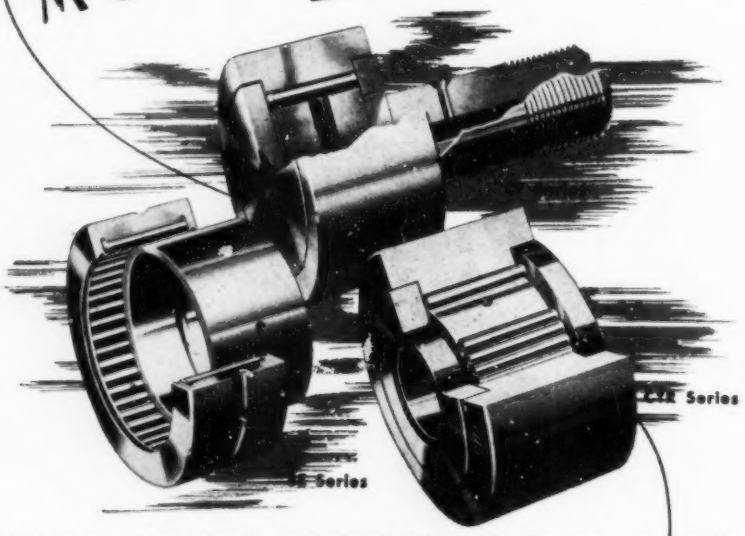

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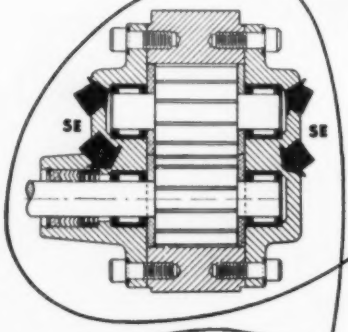

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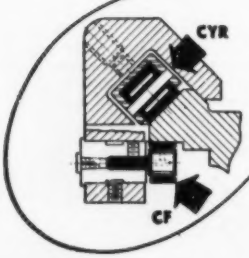

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more fundamental know-how of engineering, physics and chemistry, and if management is to make a profit and prosper while labor and material costs rise and margins shrink, more and better engineering control is paramount all along the line. This control is easiest and least painful when engineering is an inherent and integral part of all management levels and not as a policeman from a separate department.

Teachers and shoe salesmen are fine in their place, but patience wears thin when the engineer has to perennially coax, wheedle, outsmart and do end runs to make changes that should be obvious in a minute.

A little more on this theme may be helpful to industry, especially if the Korean situation snowballs into an avalanche. —H. L. RATCHFORD

Sylvania Electric Products Inc.

Ed—Mr. Ratchford's comments are also thought-provoking and may well be the basis for an editorial in a coming issue of PRODUCT ENGINEERING.

Another View Of Nodular Cast Iron

To the Editor:

This is in reference to the article on nodular cast iron by Mr. Charles K. Donoho, which appeared in the April 1950 issue of PRODUCT ENGINEERING.

In considering the use of nodular iron for machine and other parts, three questions are often raised:

1. What service improvement will be gained from its use?
2. What economic advantages will result?
3. What dependability can be expected?

These questions could be answered more satisfactorily if they were applied to individual castings where design importance of part, machining, service dependability and ultimate cost could be considered in their order of importance. However, a general answer can be reached by considering the knowledge available on nodular iron and comparing it with other materials.

What Is Nodular Iron?—Basically, this material is not new. It has the general microstructure and physical properties of malleable cast iron. Both materials differ from gray cast iron only by the form of graphite. In malleable iron, the graphite is entirely nodular, but this is not always true in the case of nodular iron It has been stated that over 25 percent of the graphite present must be nodular, the remaining 75 percent being in flake

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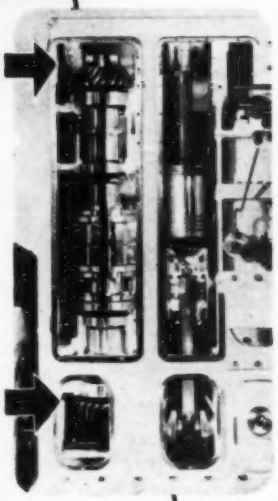

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form. Where the larger exceeds the lesser, the law of mass says that the larger must dominate. This fact certainly is true when applied to the influence of structure on the engineering properties of metals and alloys.

Must Be Fully Nodularized—A nodular casting with as much as 50 percent of the graphite in flake form will have no ductility and certainly will have no better physical properties than a good grade of cast iron. It will be vastly inferior to a steel casting. This recently was brought out in an article by Max Kuniansky, reviewing his experience at the Lynchburg Foundry Company as follows:

"For those who embrace ductile iron as a means of correcting faults in metallurgy, cupola and foundry practice, many headaches are in store. For those who sell ductile iron for any and all uses, much trouble lies ahead for both the consumer and the producer. In making pipe or castings of uniform light section that cool fairly fast, there are not many problems. For making a varied line of engineering castings, much spade work is yet to be done.

"We feel that an engineering application of ductile cast iron is from one to three years away from the first experiments with this material. Let us experiment wisely and make haste slowly."

If nodular iron is made from a soft graphitic pig iron base mix that is high in carbon and silicon and not fully nodularized, the properties quickly revert to those of a weak graphitic pig iron.

If nodularization is 100 percent, the properties show great improvement. However, if full nodularization is not obtained, then physical properties fall correspondingly as shown by the test results shown in the chart. These results were taken from 1-in. diameter test bars especially prepared. If the nodularized treatment is missed or if it is fully obtained in the lighter sections of the casting, but is not fully nodularized in the heavy sections because of process deficiency, casting design or mass influence, the casting will fail. If it is designed to the properties of 50 percent nodularity, then an unnecessarily high cost has been involved in using a material from which no advantage in service can be obtained.

Nodularizing, which is entirely a ladle treatment, sometimes is not effective although it has all the earmarks of being so. Physical properties of castings poured from the same heat can vary greatly from 50,000 psi tensile in ¾-in. sections down to 22,000 psi tensile in 4-in. sections.

Casting Soundness—Nodular cast iron exhibits high shrinkage and a

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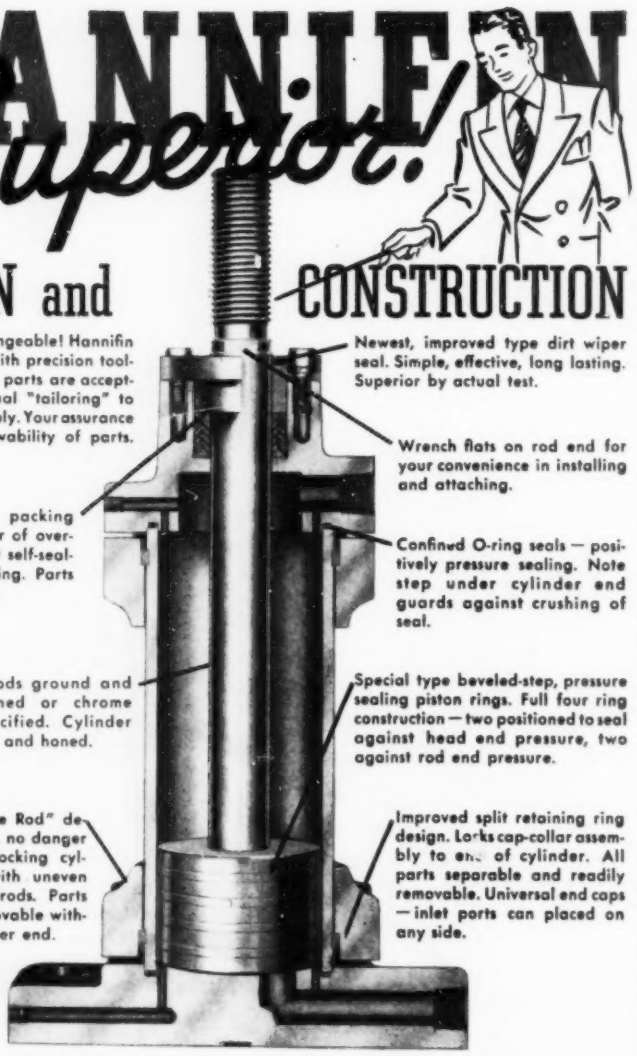

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moderate fluidity and, on casting, behaves somewhat like steel. Experimentors with the material have emphasized shrinkage defects and this is a problem that will require further study and improvements in practice.

Machining Costs—No conclusive figures are available as yet The Brinell hardness of general-run castings varies between 207 and 295. Costs therefore are likely to run higher than cast iron and will more nearly approximate those of cast steel.

Casting Costs—Commercial production has not reached the stage where dependable figures are available. The two or three foundries consulted indicate that cost per ton is over $60 more than that of cast iron. If ductility is required, the castings must be annealed at an additional cost of 2 to 5 cents per pound depending upon quantity and capital investment involved in the annealing equipment.

Further Work Needed—It appears that further development work is required before nodular iron can be classed as a fully proven engineering material, although its properties as indicated by test bars are indeed intriguing.

The successful production of nodular iron castings requires constant attention and rigid control over every step of the process. There can be no half measure because failure to attain the desired form of graphite separation may result in extremely poor physical properties. Inexact or under heat-treatment results in undesirable graphitic flake formations which, in turn, result in very low engineering characteristics. Over heat-treatment, on the other hand, will result in a brittle, non-machinable, and high-shrinking material because of the large amounts of carbides present.

The gap between experimental achievement and general commercial use has in history always been a wide one, a gap which unquestionably will be bridged through the passage of time combined with patient development and sound manufacturing procedure.

—C. E. HERRINGTON
Meehanite Metal Corp.

Material	Tensile Strength, psi	Modulus of Elasticity, $\times 10^6$	BHN
Base Iron	20,000	5	150
25 percent Nodular	43,000	14	217
50 percent Nodular	65,000	17	269
All Nodular	85,000	23	293

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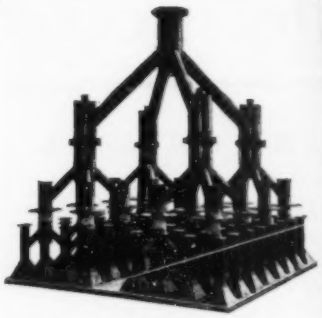
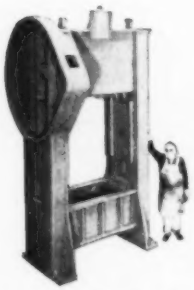
Steel-Weld FABRICATION

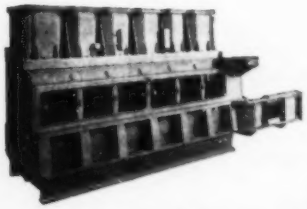


One of Several Parts of a Catalytic Cracking Plant Produced for the Petroleum Industry.

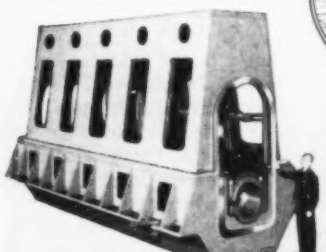
Pressure Vessels for the Chemical and Allied Industries.



Heavy Press and Machine Frames and Bases for the Machine Tool Industry.



Diesel Engine Crank Cases and Frames for the Marine and Electro-Motive Field.



Crank Cases, Frames, and other Parts for Manufacturers of Marine Steam Engines of Uniflow or Multiple Expansion Type.

Two Diesel Scavenger Housings, which were part of an order covering a number of such welded steel housings fabricated and completely machined ready for assembly by the Steel-Weld Division of The R. C. Mahon Company.

The Diesel Engine Scavenger Housings illustrated above are another example of Mahon advanced technique in Steel-Weld Fabrication . . . they are typical of thousands of Steel-Weld Fabricated parts and major assemblies produced by Mahon for manufacturers of heavy machinery and equipment throughout the country. You, may also find it advantageous to employ Steel-Weld Fabrication . . . if you do, and are interested in good workmanship, you will find in the Mahon organization an unusual source for welded steel in any form for any purpose—a source with modern fabricating and machining facilities, backed by a staff of competent design engineers and craftsmen from whom you may expect a smoother, finer appearing job, embodying every advantage of Steel-Weld Fabrication.

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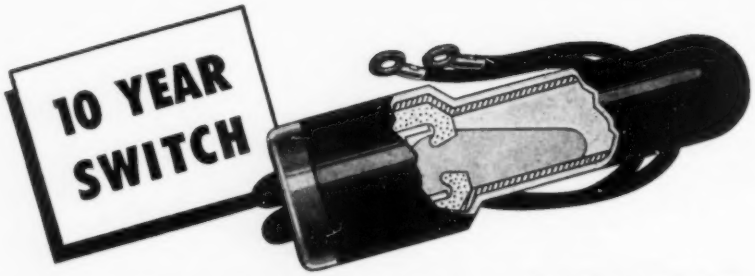

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NEW BOOKS

The Mechanical Properties Of Wood

FREDERICK F. WANGAARD, *Associate Professor of Forest Products, Yale University.* 377 *pages,* 6 x 9 *in. Published by John Wiley & Sons, Inc.,* 440 *Fourth Ave., New York* 16, *N. Y.* $6.

This volume is a completely rewritten and reworked version of its predecessor, Dean George A. Garratt's book of the same title published in 1931. Extensive developments in timber mechanics and timber engineering and increased knowledge of the effects of various factors on the properties of wood are all reflected in this book.

It is a complete source of information on the technical properties of wood. The book contains comprehensive tables presenting fundamental strength and related properties for more than 150 American woods as well as tables of recommended basic-stress and working-stress values for structural species. Examples illustrating the derivation and application of stress values are given throughout the book.

The four parts of the book are: The Basic Mechanical Properties of Clear Wood; Factors Affecting the Mechanical Properties of Wood; Working Stresses for Structural Lumber and Timber Testing. References appear at the end of each section. The Appendix contains a sample plan for testing small clear specimens of wood.

Vector and Tensor Analysis

HARRY LASS, *assistant professor of mathematics, University of Illinois.* 347 *pages,* 6 x 9 *in. Published by the McGraw-Hill Book Co., Inc.,* 330 *W. 42nd St., New York* 18, *N. Y.* $4.50.

This book is the first to be published in the "International Series in Pure and Applied Mathematics". This series is designed to meet the need for new works at the graduate and upper undergraduate level in theoretical or pure mathematics.

Methods and tools of vector and tensor analysis are developed and discussed as they are applied to geometry, mechanics, electricity, hydrodynamics and the theory of relativity.

Introductory chapters are included to familiarize the reader with the algebra and calculus of vectors. Nothing beyond calculus is required to un-

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derstand the mathematical development of the subject matter.

A simple and comprehensive outline of real variable theory, especially in its application to the Riemann integral, is included. Numerous examples are presented to illustrate the theory. Exercises at the end of each section supplement the text.

Chapter headings include: The algebra of vectors, differential vector calculus, differential geometry, integration, static and dynamic electricity, mechanics, hydrodynamics and elasticity, tensor analysis and Riemannian geometry, and further applications of tensor analysis.

Electronics, Principles And Applications

RALPH R. WRIGHT, *Associate Professor of Electrical Engineering, Virginia Polytechnic Institute,* 387 *pages,* 6 x 9 *in. Published by The Ronald Press Co.,* 15 *E. 26th St., New York* 10, *N. Y.* $5.50.

This is a textbook written primarily for non-electrical engineering students. However, it is satisfactory also for instructing electrical engineering students and physics majors.

The make-up and arrangement of the book is as follows: Chapters 1 through 3 present basic electronic principles; Chapter 4 is a brief review of d-c and a-c circuits; and Chapters 5 through 12 deal with basic applications of electron tubes. Chapter 4 has been introduced to adapt the text to students of various backgrounds. Although one or more elementary courses in electrical engineering are ordinarily prerequisites for electronics, many students require a brief review of basic circuit theory, as presented in this chapter, before taking up analysis of electronic circuits. If the instructor prefers to leave to the student the responsibility of review, omission of Chapter 4 will not affect the continuity of the text.

Chapters 5 through 12, devoted to electronic circuits, have also been planned with flexibility in mind. They are arranged in the book in an order which the author considers a logical one, though it is by no means the only desirable order of coverage. Each of these chapters is more or less a complete unit in itself; thus the actual order of use may be any that seems most suitable.

Every effort has been made to present the material in a clear and coherent manner, slanted to meet the requirements of nonelectrical students. Ordinarily nonelectrical students will

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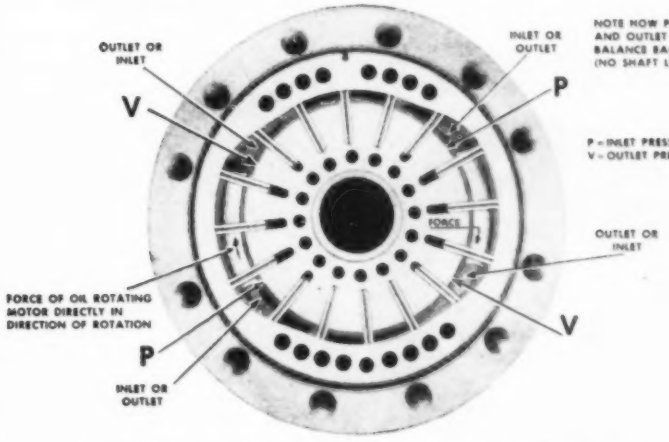

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RAW MATERIAL

THE CUSTOM MOLDER HAS LONG SINCE resigned himself to his allotted position in the plastics industry, tucked cosily behind the eight ball. It does have the doubtful advantage of being shady there. This will not stop him from letting out with occasional cries of pain when someone rolls the aforesaid eight ball over on his toes without warning. He doesn't like to be disturbed at his efforts to make a meal from the incautious items which flit in his direction.

The ebb and flow of plastics raw materials are a matter of real concern to him. At times this concern rises to wonderment.

Case history No. 1—Phenolics . . . During the war, many molders had to shift their sources of supply from the accustomed channels. After the war, phenolics were scarce and went on allotment. Some makers set the allotment on pre-war purchases, some on war purchases. Result—headaches for the unlucky ones.

Case history No. 2—Polyethylene . . . This was the darling of radar. When wifely intuition proved scientifically accurate in plotting the whereabouts of her wandering spouse, it was discovered that the demand for this plastic was non-existent. So the R.M.M. (Raw Material Makers) called on the molders to find uses. That takes time, so the R.M.M. were probing too. Result—1948, supply plentiful; 1949, on allocation; 1950 (first quarter), supply plentiful; 1950 (second quarter), on allocation.

Case history No. 3—Polystyrene . . . This one really speeds up the confusion. January 1950, overnight delivery; April 1950, pending price war; May 1950, perhaps 60 days.

All the poor custom molder has to do is convince his customers that they should order molded parts three months ahead and that their competitors are, to put it mildly, exaggerating if they claim to have an inside track and can get material any time they order it.

We are used to our position relative to the aforesaid eight ball, but we get confused when someone pushes it over on us without fair warning.

GEORGE K. SCRIBNER, *President*

Boonton Molding Company, Boonton, N. J.

not require an intimate knowledge of circuits. Rather, they need a good grounding in basic principles of electronics and an understanding of how these principles can be applied to control or facilitate various industrial processes. In consequence, a highly mathematical approach has been deemed undesirable. Wherever possible topics are presented from a qualitative rather than a quantitative point of view. Only basic equations have been included. In most instances these are not derived; they are simply stated with the terms in each equation explained sufficiently to enable the student to use them intelligently. For the most part, only elementary mathematics is used, with employment of simple calculus in a few places to analyze phenomena that cannot be readily analyzed in any other way. In Chapter 12, which deals with control circuits, no attempt has been made to describe specific control equipment. Instead, various types of basic circuits illustrating the principles of electronic control circuits are explained.

Quality Control and Statistical Methods

EDWARD M. SCHROCK, *General Electric Company, 213 pages, 6 x 9 in. Published by Reinhold Publishing Corp., 330 W. 42 St., New York, N. Y.* $5.

In straight-forward language, this book follows the step-by-step process through the operations of applying statistical methods to the problems of modern industrial quality control.

Many illustrations and practical examples reveal the use of the newest techniques to achieve almost unbelievable improvements in quality uniformity and reduction of scrap and rework losses. This book is intended for those who are new to the field who want to improve their effectiveness in appraising and controlling the quality of products or the economy of methods to increase production.

The presentation and treatment of materials is in accordance with generally accepted practices and standards. The application of methods covers the broad field of industry and includes such diverse problems as are found in the electrical industry, growing and packaging of food, steel products, errors in filing, mail orders, clothing, lumber, controlling overtime, optical instruments, biologicals, textiles, and polls of public opinion. In short, this book involves any field where the quality of product must be appraised and controlled and includes numerous

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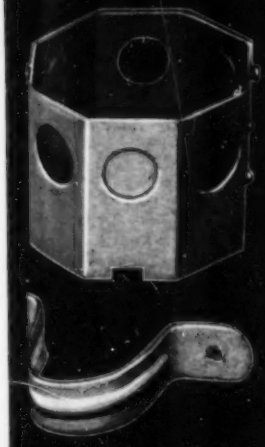

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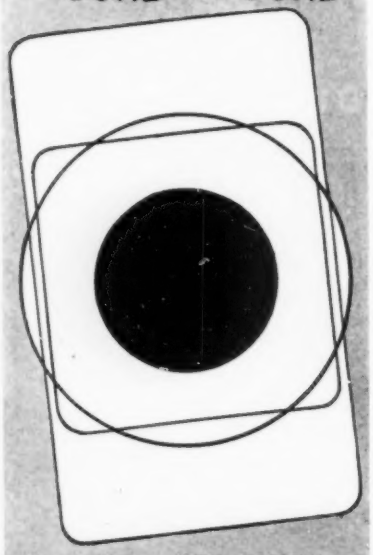

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topics, materials and useful tables not previously covered.

The organization of this book makes it suited for use in classroom instruction in the college field. It presents a logical, practical approach to those who are new to the subject. Its comprehensive treatment of widely useful techniques provides an excellent foundation for those who wish to specialize in the field. There is sufficient material for a full yea. course on industrial quality control.

Introduction To The Transfer Of Heat and Mass

E. R. G. ECKERT, *Air Materiel Command, Wright Patterson Air Force Base; and consultant, National Advisory Committee for Aeronautics.* 284 *pages,* 6 x 9 *in. Published by the McGraw-Hill Book Co., Inc.,* 330 *W.* 42*nd St., New York* 18, *N. Y.* $4.

Written at the graduate level, this translation and adaptation of the author's German text will be of interest to mechanical engineers.

The volume is designed to give an understanding of the basic physical processes and the different modes of heat transfer, and the close relationship between the transfer of heat and mass. The evaporation and condensation of water vapor is used as the basis for the treatment of heat transfer. Understanding these basic processes will enable the reader to use the formulas for the calculation of heat exchanges and cooling problems.

New topics such as regenerative heat exchangers, heat transfer at high velocities, and studies with the zehndermach interferometer are included.

The chapter headings include: Fundamental principles of heat transfer, heat conduction, heat transfer by convection, thermal radiation and the exchange of mass.

In the appendix of 18 pages, which is intended primarily to support the text and not to fulfill the requirements of a handbook, tables are presented that give the property values of the important liquids, gases, metals and nonmetallic materials.

Industrial Instrumentation

DONALD P. ECKMAN, *College of Engineering, Cornell University.* 396 *pages,* 5½ x 8½ *in. Published by John Wiley & Sons, Inc.,* 440 *Fourth Ave., New York* 16, *N. Y.* $5.

This book is an introduction to the science of measurement. Rapid development of new industrial methods and

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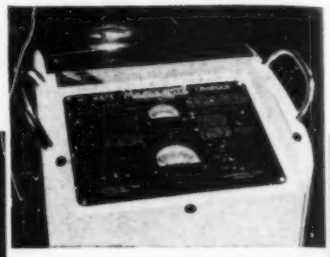

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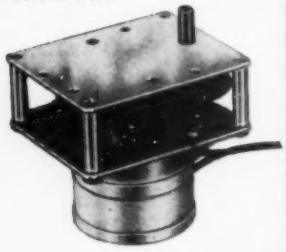

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techniques has brought the subject of measurement into a science on its own physical foundation and has extended its scope far beyond the elementary concepts of temperature and pressure determinations. New measurement methods based on both new and old physical phenomena are becoming common in industrial operations.

The general aim of this book is to review the principles of those methods of measurement used in industrial processing and manufacturing. Primary emphasis is given to the method rather than to the mechanism. Attention is paid, however, to important practical details such as the selection and arrangement of instrument, process analysis and supervision of instrument equipment. Composition analysis instruments and methods are treated in a chapter that covers emission and absorption spectroscopic analysis, mass spectrographic analysis, and other important methods. Instruments for mechanical measurements including displacement gages, strain gages, force meters, velocimeters, and accelerometers are treated in another chapter.

Numerous schematic diagrams of operation principles and meter installations are used. A 30-page appendix contains temperature conversion tables, standard thermocouple emf tables and orifice flow coefficient tables.

Transient Performance Of Electric Power Systems

REINHOLD RUDENBERG, GORDON MCKAY *Professor of Electrical Engineering, Harvard University.* 832 *pages,* 6¾ x 9¼ *in. Published by the McGraw-Hill Book Co., Inc.,* 330 *W.* 42*nd St., New York* 18, *N. Y.* $12.

The field of transient electric phenomena has developed into an important foundation of electrical engineering. For the design and the operation of electrical systems it has assumed the same significance as the regular performance in the steady state. Moreover, some of the accidental transient phenomena, like short circuits in a network, instability of the generating machinery, and lightning disturbances on transmission lines, have become of such outstanding importance that the layout of modern power systems is predominantly determined by their conditions.

We consider as transient performance the reaction of electric circuits and their associated machines and apparatus to any nonstationary influence. It does not matter, in principle, whether such perturbation is inten-

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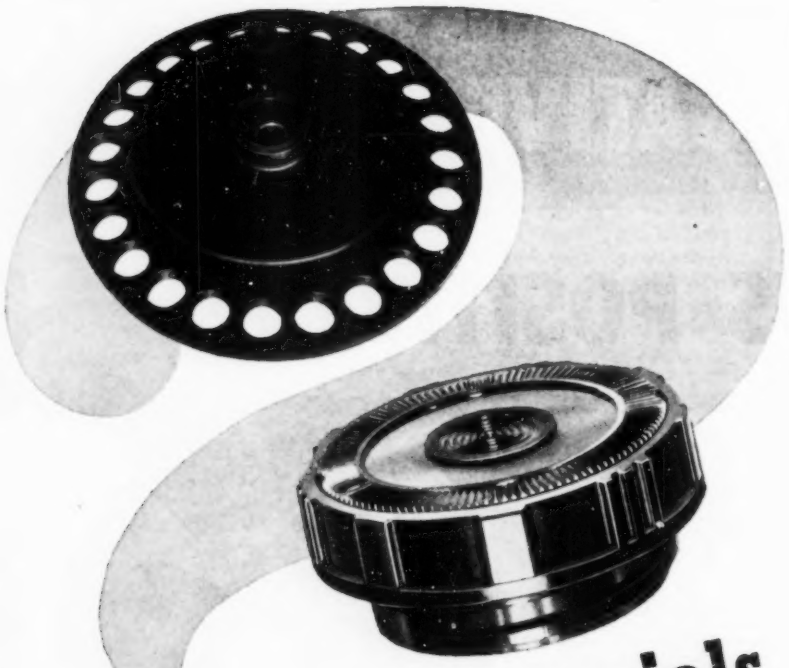

What materials would you choose?

1. CHLORINE CELL COVER: Previous cement covers deteriorated rapidly, gave rise to contamination in vorce cell, with consequent increased current and maintenance costs. Now made of material that combines high strength with superior electrical and chemical resistance to give longer life, cleaner product. What material?

2. BATTERY VENT CAP: Permits gases to escape from auto battery without letting acids out. Must be low in cost, resistant to acids. Should permit high molding rates, easy assembly and cementing of two halves. What material would you try first?

Answers: No. 1—Ace Molded Hard Rubber greatly increased life of vorce cell covers, may do same for your designs. No. 2—Try Ace Parsan (polystyrene) *first* for any job similar to this Ace-molded vent cap.

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tional or accidental, or whether the transient behavior will lead the system to a new state of regular operation or a distorted condition, as often connected with excess currents or voltages.

The book deals with the transient performance of power circuits with lumped properties. This is an enlarged edition in English of this classic German text. The material is arranged so that it progresses in all the main sections from the simpler to the more difficult problems. Mathematical analysis is considered as an auxiliary tool, and emphasis has been placed on the physical understanding of the analysis of an example and the ease of discussion of the result rather than on the mathematical rigor and generality of the method of solution. Many numerical examples, taken from actual practice, are worked into the text. The numerous oscillograms and measurements throughout may further bridge the gap between the theory and the practice of transients.

The book tends to develop a survey on the entire field of transients in lumped power circuits. The various chapters considering different phenomena are built up as independently as possible, with minimum citations of preceding material. For simplification, the equations and figures are numbered anew in every chapter.

High-Frequency Induction Heating

FRANK W. CURTIS, *Consulting Manufacturing Engineer.* 389 *pages,* 5½ x 8 *in. Second edition, published by the McGraw-Hill Book Co., Inc.,* 330 *W. 42nd St., New York* 18, *N. Y.* $6.

The purpose of this book is to offer the user of high-frequency induction-heating equipment basic technical application details that will serve as a ready reference in connection with the heating of metal parts, such as may be required for hardening, heat-treating, brazing, soldering, annealing, stress-relieving, forging, upsetting or melting. Though the generation of high-frequency current is an electrical function, putting this current to practical use becomes almost entirely mechanical. Therefore, other than brief references to electrical data characteristic to this type of heating, the text is confined to application technique.

Here is a practical explanation of the increasingly important technique of applying heat in the production of metal parts. This up-to-date book explains the fundamental electrical principles involved in high-frequency in-

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Alloy: **302 Stainless Steel**

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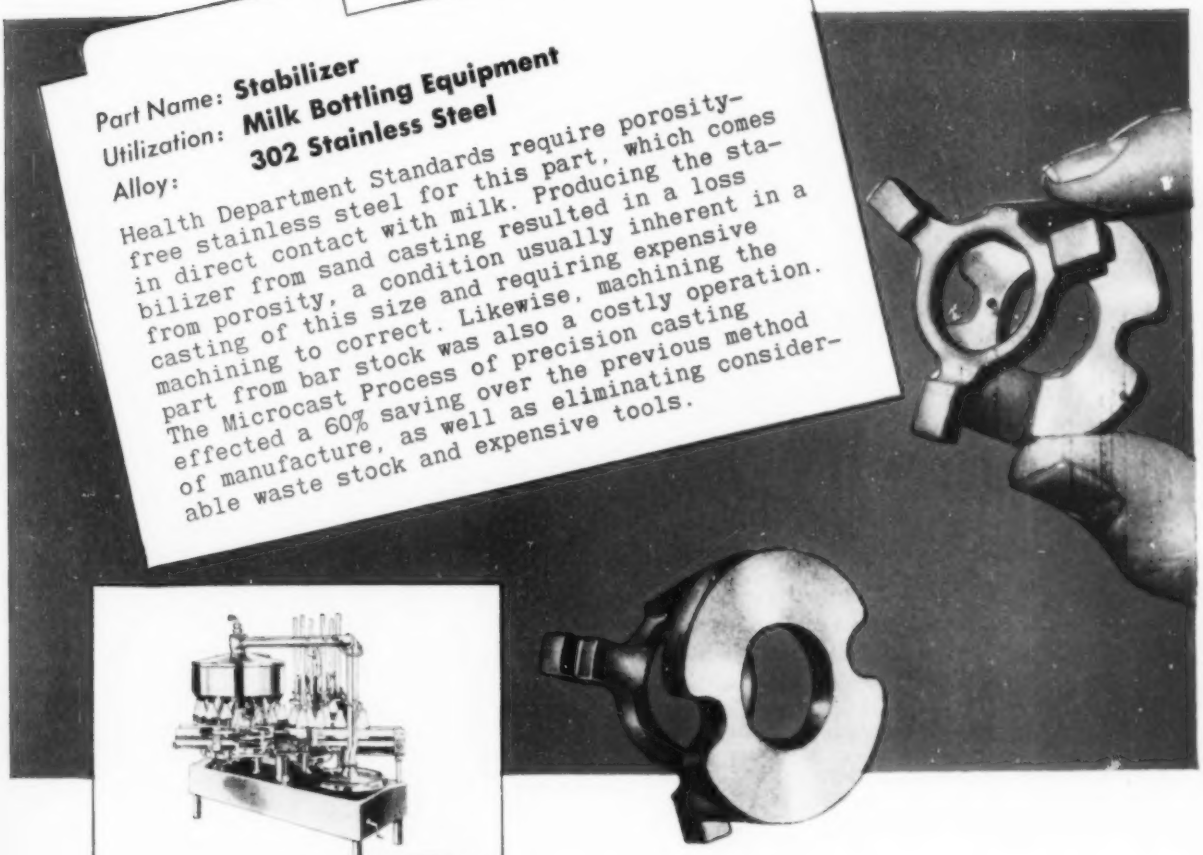

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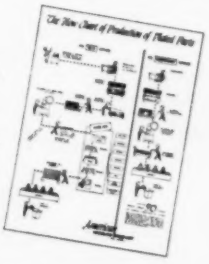

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duction heating; defines its range and scope; describes construction design of induction-heating coils; and discusses in detail many applications of induction heating to specific heating problems—for hardening metal parts, joining of metal assemblies and similar operations.

This revised second edition includes a comprehensive analysis of coil design and timely new material on fixture designs. Types of steel best suited to induction hardening, various means of quenching, soft solders, brazing alloys, and fluxes are all treated in full.

Properties of Metals at Elevated Temperatures

GEORGE V. SMITH, *Research Laboratories, U. S. Steel Corp., Kearney, N. J.* 401 pages, 6 x 9 *in. Published by the McGraw-Hill Book Co., 330 W. 42nd St., New York, 18, N. Y.* $7.

The fact that our industrial progress has been predicated on the metals and materials that make up machines is true today as ever. For although thermodynamic calculations indicate high efficiencies and greater possible power, the problem remains one of finding materials that will operate satisfactorily under extreme temperatures. The research work that is being conducted on internal combustion engines for high-speed aircraft, power plant design and nuclear engineering is aimed at seeking new developments in the materials for high temperature use. This book brings together and correlates the results of research work that has been carried out on metals at elevated temperatures during the past twenty-five years.

As a comprehensive summary of the available knowledge on this subject, the book should be useful to many engineers and particularly metallurgists whose responsibility is that of choosing metals for elevated temperature service. Most of the material is of a metallurgical nature. There are chapters on: Plastic deformation of single crystals and polycrystalline metals; creep of polycrystalline metals; theories of flow and fracture; and effect of chemical composition on creep.

Design engineers should find the chapters on effects of varying temperature and load, changes during service and working stresses of particular interest.

The Appendix discusses the super alloys that are available in this country and Great Britain. There is a very complete bibliography of 547 references for those who might be interested in further study of the subject.

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